COMBUSTION

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CHICAGO DISTRICT ELEC. GENERATING CO., HAMMOND, IND.—CHICAGO AERIAL SURVEY

Application of the Second Law of Thermodynamics to Power Plants and Industrial Processes

By WM. L. De BAUFRE

Determination of the Economical Size of a Natural Draft Chimney

By J. G. MINGLE

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A. R. SMITH • A. R. MOBERG • PAUL E. ROLL • DAVID BROWNLIE • B. J. CROSS

COMBUSTION

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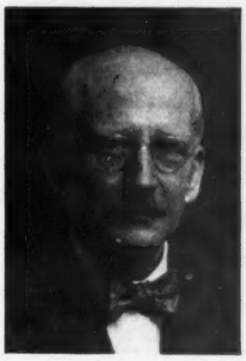
VOLUME 2 . DECEMBER 1930 . NUMBER 6

High Pressures and High Temperatures



GEORGE A. ORROK

IT is now more than five years since the first 1,200 lb. installation was put in service at the Edgar Station of the Edison Illuminating Company of Boston. Since that time, the Lakeside installation at Milwaukee, the Northeast installation at Kansas City, the Holland, Deepwater and South Amboy Stations in New Jersey have been put into service, while the San Francisco plant and the plant of the Ford Company at Fordson are nearly ready to commence work. These stations are all 1,200 to 1,400 lb. stations with 750 deg. fahr. steam temperature. The designs cover a wide range and the operating conditions have shown no serious troubles which would not have been encountered if ordinary pressures had been used. The trend of prices shows that the costs of these stations are little, if any, in excess of plants built for ordinary pressures.

In Europe, total steam temperatures of 850 deg. fahr. are common; at least three stations are running at that temperature, while the Witkowitz plant has been running now for more than two years at 950 deg. fahr. Reports from all these plants agree that the troubles have been those which usually occur with the lower pressure plants. Economies have been most satisfactory. It would appear that pressures of this general order with steam temperatures between 750 and 950 deg. fahr. with the use of reheat has become a standard type of installation.

The Benson system running at or about the critical pressure, and the Loeffler system, used at Witkowitz at about 1800 lb., are far along in the experimental stage and the profession is eagerly awaiting operating reports from Langerbrugge where the newest and largest Benson installation is now in service. Meanwhile, a number of most interesting plants have been installed in the 400 to 750 lb. range, in which reheating has not been necessary, with economies of a most excellent order. Operating results attained by these plants have been most satisfactory. The trend at the present time is uncertain but it would appear that both ranges, 400 to 750 lb. and 1,200 to 1,800 lb., will be used in succeeding designs, while the critical pressure type will continue to be most interesting.

The development of the standard types of boiler for pressures as high as 1,800 lb. necessitating forged or welded drums has been successful, while the Benson and Loeffler types require no large size drums. Other drumless types of boilers are being considered and the engineer has the same variety of choice for the higher pressures as he has in the lower range.

We are approaching the optimum economy in the use of heat in a steam power plant and it is not probable that major savings will be made without the discovery of some essentially new principle.

Geo A Orrok

Consulting Engineer, New York

EDITORIAL

Why Eulogize Ignorance?

"SOMETIMES, I have wondered if education does not hamper the pioneer. It is so full of warnings not to make the attempt. Ignorance may sometimes step in where wisdom fears to tread."

The Engineer of London thus quotes from an address recently given by a leading engineer of Great Britain.

If this statement typifies British engineering thought, or, if it represents the consensus of any particular engineering group; in fact, if it is anything more than an isolated personal opinion—it merits refutation.

The cumulative efforts of centuries have brought to modern engineering a rich heritage of experience. Education provides for the study and recapitulation of these experiences of the past and in the process we acquire knowledge which enables us better to build for the needs of the future.

Ignorance, on the other hand, discounts all traditional experience and starts each generation at scratch—no further ahead than its predecessors.

Perhaps long years ago, when ignorance was common, ignorance was no serious handicap to advancement, but today, when knowledge is universal, knowledge is indispensable to progress.

Three hundred years ago, when Galileo first swept the heavens with his telescope, each new position of the instrument brought countless stars into vision for the first time.

Today the star fields of lesser magnitude have all been surveyed, and he who would find new stars must first equip himself with a telescope of sufficient strength to reach beyond the limitations of the past.

When the sum total of human knowledge was small, men on blind errands occasionally blundered onto worth while achievements.

Thus, Columbus sailed forth seeking the Indies and discovered America, the age old search for perpetual motion yielded many ingenious mechanical movements, while the alchemists in their untiring efforts to transmute base metals into gold and silver, compounded an endless array of valuable alloys. These pioneers, living in an era characterized by ignorance, found ignorance no hardship.

The outstanding physicist of two centuries ago climaxed his work by defining the law of gravitation. This is where the modern technical student begins. The average boy studying physics in high school has a better basic knowledge of electricity than had the college graduate of a generation ago.

Life today moves at a rapid tempo.

Modern engineering is a complex structure that has long since passed the stage where mere native ingenuity and digital dexterity can contribute appreciably to its advancement.

The modern pioneer who seeks to raise the standards of engineering must bring to his work the best of modern tools—chemistry, metallurgy, physics and mathematics—and the ability to use them intelligently. He must be adept at research and capable of translating test results into rational relationships so that he may apply his knowledge of known applications to the solving of new problems.

The span of human life is too short to permit the devoting of any time to duplicating the mistakes of the past. The blind blundering of ignorance inevitably leads to wasted time, wasted money and wasted opportunity.

Knowledge not only is power—it is the basis of progress. Engineering advancement can no longer be whittled out with a jack-knife.

Association Work— An Obligation, An Opportunity

GOETHE, the great German author and poet, once said that the sources of his thought were so numerous that one who traced them out would find it difficult to attribute any originality to him. Here was a man whose genius and creative powers won him universal recognition and yet he felt compelled to admit that his originality was second-hand,—that it was derived from the accumulated knowledge and thought of preceding centuries.

Man of himself is futile. His power of accomplishment is the result of his ability to understand and use the knowledge which is the heritage of all humanity. It is the development of this ability, enabling the individual to take advantage of all that others have learned about the things in which he is interested, that has made possible the rapid progress of recent centuries in all fields of human endeavor.

Today the interchange of knowledge and experience has become a highly developed technique. Leaders in all fields of thought have learned the value of cooperation and collaboration.

No better instrumentality is provided for extending the usefulness of up-to-the minute information than our various societies and associations.

Every engineer should regard association activity as an important part of his work. The heads of organizations should encourage their employees to attend association meetings and contribute of their knowledge and experience. Those who follow this policy will receive as much, or more, than they give. The benefits are mutual and far reaching.

Determination of the Economical Size of a Natural Draft Chimney*

By J. G. MINGLE
Indianapolis, Ind.

The author points out the fallacy of the practice, all too frequently followed, of basing chimney sizes upon boiler horsepower. He then discusses the factors upon which such size determinations should be based and presents equations for arriving at the correct diameter and height of stack for the particular conditions to be met. Emphasis is placed on the importance of gas velocity as a factor having a considerable bearing on chimney cost. In order to demonstrate the use of the methods and equations presented, a hypothetical example is given and worked out.

THE chimney in a steam boiler plant is one of the most important parts of the entire installation, yet, as a general rule, very little attention has been paid either to determining its proper, and also economical, size or to its relation to the other parts of the plant. The size of the great majority of chimneys which have been built in the past has been determined by the aid of a table of chimney sizes based on boiler horsepower. After the maximum, or ultimate, horsepower of the boilers had been ascertained, the size in the table corresponding to this figure was then selected. Generally, no further attempt was made to investigate the performance of the size selected to determine if the height was sufficient to create the required amount of draft necessary to overcome the various losses of draft throughout the installation outside of the chimney itself and also if the diameter, or area, was sufficient to enable the structure to quickly and economically take the gases away from the furnace and discharge them to the atmosphere. It is the purpose of this article to develop a system of determining the economical size of a natural draft chimney which will take into consideration all of the operating conditions to which the chimney will be subjected.

In designing a new chimney of the natural draft type, it must be decided, first of all, whether the chimney is to be the sole producer of draft, or whether some auxiliary, such as a blower or an induced draft fan, is to be used in the production of the required amount of draft demanded by the operating plant. It has become the usual procedure, in designing central station and other large plants, to use a combination of a chimney and a fan to produce the required draft: the chimney being of sufficient size to produce the necessary draft to overcome the resistances, or losses of draft, through the boiler and accessories and the fan of sufficient size to produce the draft to overcome the resistances through the pit, grates and fuel bed, the capacity in both cases being such that either type of pressure transformer will handle the gases from the boilers when they are operated at their maximum rating. In the great majority of the plants where chimneys are used at all, they are used to the exclusion of all the other types of draft production and create the entire amount of draft necessary to overcome all of the resistances encountered throughout the entire plant.

In general, the operating conditions which must be considered in designing a natural draft chimney may be summarized as follows:

1. Atmospheric temperature
2. Chimney gas temperature
3. Chimney gas density
4. Altitude of plant
5. Required draft consisting of:
   a. Draft loss through the fuel bed and grates
   b. Draft loss through the boiler
   c. Draft loss through the accessories
   d. Draft loss through the breeching
6. Amount of gases to be handled by the chimney.

Speaking in general terms, the height of a properly designed natural draft chimney is dependent upon the amount of draft required to overcome the various resistances offered to the flow of gases throughout the entire boiler plant installation, or such part of it which the chimney is to serve, and also the temperature differential between the chimney gases and the atmosphere. The area, or diameter, is dependent upon the amount of gases generated and passing through the installation and also the temperature and velocity of these gases. The atmospheric pressure corresponding to the altitude and the density of the gases affect the size to a relatively small extent. The various losses of draft throughout the installation vary as the square of the velocity of the gases and, hence, the height is also affected by the chimney gas velocity.

Based on an analysis of the performance of a natural draft chimney, the equations for the height and diameter may be stated as follows:

$$D = 0.288 \sqrt{\frac{W\,T_c}{W_c\,B_o\,V}} \qquad (1)$$

$$H = \frac{D_r + \dfrac{0.046\,W_c\,B_o\,V^2}{T_c}}{2.95\,B_o\left(\dfrac{W_o}{T_o} - \dfrac{W_c}{T_c}\right) - \dfrac{0.184\,f\,W_c\,B_o\,V^2}{T_c\,D}} \qquad (2)$$

in which:

D = minimum diameter, ft. (most chimneys are built circular in section)

D_r = Required draft, in. of water

H = required height above the grate bars, ft.

V = chimney gas velocity, ft. per sec.

W = amount of gases generated and passing through the chimney, lb. per sec.

T_c = chimney gas temperature, deg. abs.

T_o = atmospheric temperature, deg. abs.

B_o = atmospheric pressure corresponding to altitude, in. of Hg.

W_o and W_c = density of the atmosphere and chimney gases, respectively, at 0 deg. fahr. and sea level atmospheric pressure, and f = coefficient of friction.

These two equations may be simplified as follows:

$$D = K_d \sqrt{\frac{1}{V}} \qquad (3)$$

$$\text{and } H = \frac{D_r + K_a V^2}{K_b - \dfrac{4\,K_c\,V^2}{D}} \qquad (4)$$

in which K_a, K_b, K_c and K_d = constants representing the various combinations of values of the chimney gas temperature, atmospheric temperature, chimney gas density and atmospheric pressure corresponding to altitude, among others.

Reference to these two equations will disclose the fact that the velocity appears as a positive term in the numerator and as a negative term in the denominator of the equation for the required height, and as

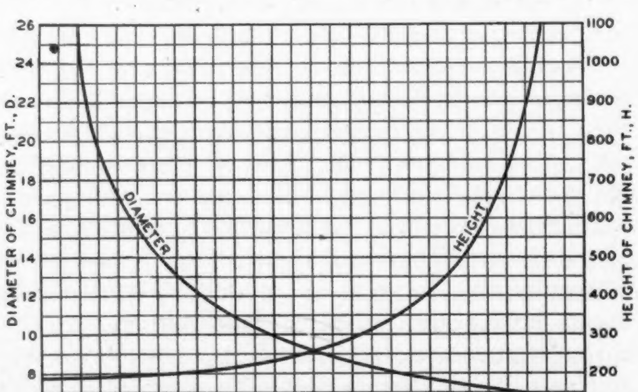


Fig. 1—Variation in diameter and height for different gas velocities, based on operating conditions given in example.

a positive one-half power term in the equation for the diameter. Hence, it is quite apparent that the height will be comparatively low and the diameter comparatively large when a relatively low value for the velocity is assumed. On the other hand, when a relatively high value for the velocity is assumed, the height will be comparatively great and the diameter comparatively small. Any value for the velocity, reasonably assumed, will result in a size of chimney which will answer the purpose and fulfill all requirements, provided all of the other factors appearing in both equations have been taken into consideration.

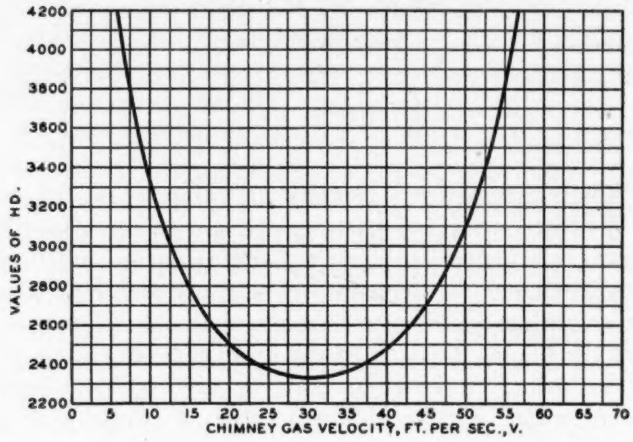


Fig. 2—Variation in values of HD for different gas velocities.

All of the factors appearing in equations 1 and 2, with the exception of the chimney gas velocity, can be either observed or values can be computed for them to a fairly close degree of accuracy. In order to arrive at some definite size, it is, of course, necessary to assume an arbitrary value for the velocity. From a theoretical standpoint, there are as many different sizes of chimney, that is to say combinations of height and diameter, for any one set of operating factors as there are values of velocity to be assumed. In order to limit the range of choice in the selection of a value for the velocity, it is necessary to establish some method or criterion as to the cost of the chimney structure, since the relative cost should determine which size of any number of choices will be built. It is quite obvious that, all other factors being equal, a short chimney will cost less than a tall one, and, likewise, one with a small diameter will cost less than one with a large diameter, speaking in relative terms in each case. Now the selection of the relatively cheapest structure becomes quite easy when the choice is limited to any number of heights with the same diameter, or to any number of diameters with the same height. However, when there are several different heights, each with a different diameter, the selection of the relatively cheapest size is by no means easy and the task becomes increasingly difficult as the number of proper sizes is increased.

The volume of material in the shaft of a chimney structure is given by the approximate equation:

$$Q_c = \pi\,D\,t\,H \qquad (5)$$

in which Q_c = volume of shaft

D = median diameter

t = wall thickness

H = length of shaft

For all practical purposes, the median diameter may be assumed to be equal to the inside diameter of the chimney. The value of t will not vary appreciably within the range of sizes for any one set of operating conditions, provided the velocities are reasonably assumed. Hence, for all practical purposes, the value of πt will be constant regardless of the size. Therefore, in general, the volume, and consequently the cost, may be based on the factor HD as a criterion.

The economical designing velocity of the chimney gases is that velocity which will result in a combination of a height and a corresponding diameter of a chimney whose cost will be less than that of any other combination of height and diameter as determined by any other velocity. From the foregoing, it will be seen that the relatively cheapest structure and also the least velocity will be attained when the height is least. Likewise, the relatively cheapest structure and the greatest velocity will be attained when the diameter is least. Stated in other words, a comparatively low velocity will insure a relatively cheap structure insofar as the height is concerned while a comparatively high velocity is required to insure a relatively cheap structure insofar as the diameter is concerned. Hence, it is necessary to find such a velocity for any one set of operating conditions as will result in a structure whose height will not be too great nor the diameter too great and at the same time result in the relatively cheapest structure.

Therefore, the value of the chimney gas velocity which will result in the least value of HD will produce a structure whose cost will be least and, as a result, will be the most economical velocity to use.

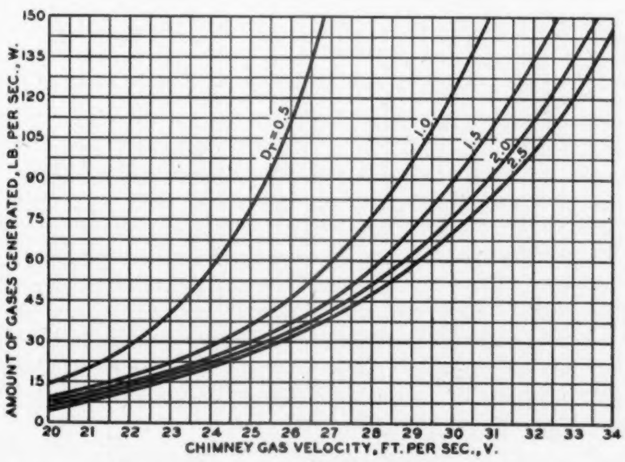


Fig. 3—Values of V_e for various values of W and D_r as determined from equation 8.

The problem at hand is to deduce an equation for the chimney gas velocity which will result in a combination of height and diameter whose product, HD, will be least.

Substituting equation 3 in equation 4 and reducing and then taking the product of the two:

$$HD = \frac{D_r K_d^2 + K_a K_d^2 V^2}{K_b K_d V - 4 K_c V^3} \qquad (6)$$

Equation 6 gives the value of HD in terms of V and a series of constants representing combinations of various operating factors. When proper values for these constants have been determined and the results for the equation plotted, it will be found that the value for HD will gradually decrease as the assumed chimney gas velocity increases until a certain minimum has been reached after which the value of HD

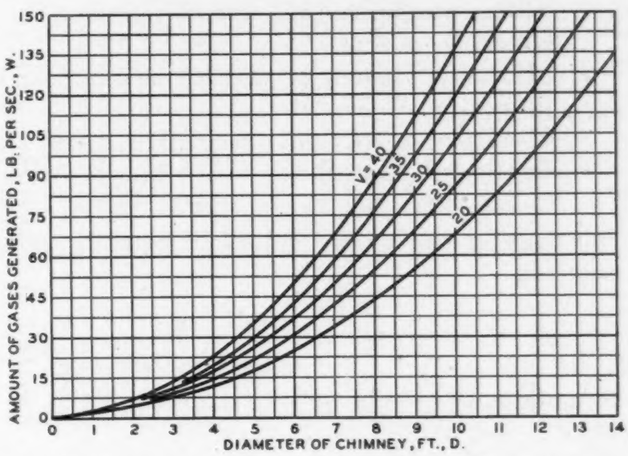


Fig. 4—Values of D for various values of W and V_e as determined from equation 9.

will increase. There will then be a minimum value for HD for a certain value of V. (See Fig. 2). The value of V corresponding to the minimum HD is the economical designing velocity. This velocity will result in a size of chimney whose value of HD will be least, and whose cost will be less than that of any other size as determined by any other velocity.

The equation for the value of V for a minimum HD is found by differentiating equation 6 with respect to V and equating the resulting equation to zero. After this has been done and the proper factors substituted for the various K constants, the following equation results:

$$V = V_e = 2.7\sqrt{\dfrac{1}{\dfrac{9.4\, f\, W_c\, V^{\frac{1}{2}}}{T_c\left(\dfrac{W_o}{T_o} - \dfrac{W_c}{T_c}\right)\sqrt{\dfrac{W\ T_c}{W_c\, B_o}}} + \dfrac{W_c\, B_o}{T_c\, D_r}}} \qquad (7)$$

in which V_e = economical designing velocity, ft. per sec.

This equation is indeterminate in that it contains the factor V whose value is being sought. A fairly close approximation for a trial value may be had by assuming $V^{\frac{1}{2}} = 30^{\frac{1}{2}} = 5.5$ after which another trial value corresponding to the value obtained by the first trial will usually result in the equation being satisfied.

After the value of the economical designing velocity has been found, the economical height and diameter may be found by substituting this value, among the others representing the operating conditions, in equations 1 and 2.

Example. Determine the economical size of a natural

draft chimney for the following operating conditions:

1. Average chimney gas temperature, 500 deg. fahr. $T_c = 960$
2. Mean atmospheric temperature, 62 deg. fahr. $T_o = 522$
3. Sea level atmospheric conditions. $B_o = 29.92$
4. Chimney gas density, 0.09 lb. per cu. ft. at 0 deg. fahr. and $B_o = 29.92$ $W_c = 0.09$
5. Total draft loss through the fuel bed, grates, boiler, accessories and breeching, 1.25 in. of water. $D_r = 1.25$
6. Maximum amount of gases to be handled, 100 lb. per sec. $W = 100$
7. Coefficient of friction, 0.0.016. $f = 0.016$
8. Density of atmosphere, 0.0863 lb. per cu. ft at 0 deg. fahr. and $B_o = 29.92$. $W_o = 0.0863$

Substituting the proper values in equation 7 and reducing:

$$V_e = 2.7\sqrt{\frac{1}{0.00104\ V^{\frac{1}{2}} + 0.00224}}$$

Assuming $V^{\frac{1}{2}} = 30^{\frac{1}{2}} = 5.5$, then:

$$V_e = 2.7\sqrt{\frac{1}{0.00571 + 0.00224}} = 30.4$$

The assumed value for $V^{\frac{1}{2}}$ was fairly close to the real value and the economical velocity for the operating conditions noted, therefore, is 30.4 ft. per sec.

Substituting the proper values in equation 1, then for an economical designing velocity of 30.4 ft. per sec.:

$$D = 0.288\sqrt{\frac{96000}{82}} = 9.9$$

Finally, substituting the proper values in equation 2, then for an economical designing velocity of 30.4 ft. per sec.:

$$H = \frac{1.25 + 0.12}{0.00631 - 0.00077} = 248$$

Hence, the required size of chimney required for the operating conditions noted, based on an economical designing velocity of 30.4 ft. per sec., is 248 ft. × 9.9 ft., or practically 250 ft. × 10 ft.

Fig. 1 shows the variation in both the diameter and the height for various velocities for the operating conditions noted in the example and Fig. 2, the variation in the values of HD for the corresponding velocities. These figures graphically illustrate the theoretical matter developed in the text and disclose the great variation in the size of chimney for any one set of operating conditions.

Equations 1, 2 and 7 appear, at first sight, to be rather cumbersome to handle but the numerical reductions, after the substitutions have been made, are comparatively simple and can be done with ease and facility with the aid of a slide rule.

These three equations, however, may be considerably simplified for general use by assuming the following general average operating conditions:

Mean atmospheric temperature of 62 deg. fahr. $T_o = 522$

Average chimney gas temperature of 500 deg. fahr. $T_c = 960$

Sea level atmospheric pressure of 29.92 in. of Hg. $B_o = 29.92$

Constant coefficient of friction of 0.016. $f = 0.016$

Chimney gas density of 0.09 at 0 deg. fahr. and $B_o = 29.92$ $W_c = 0.09$

Substituting these values in equations 1, 2 and 7 and reducing:

$$V_e = 51\sqrt{\frac{1}{\frac{3.8\ V^{\frac{1}{2}}}{W} + \frac{1}{D_r}}} \qquad (8)$$

$$D = 5.44\sqrt{\frac{W}{V_e}} \qquad (9)$$

$$H = \frac{D_r + 0.000129\ V_e^2}{0.00634 - \frac{0.00000827\ V_e^2}{D}} \qquad (10)$$

Fig. 3 gives the values of V_e for various values of W and D_r as determined from equation 8: Fig. 4, values of D for various values of W and V_e as determined from equation 9: and Fig. 5, values of H for various values of D_r and V_e for a constant diameter of 10 ft. as determined from equation 10. These three equations and figures should be used cautiously inasmuch as they give results which are true only for the

(*Continued on page 40*)

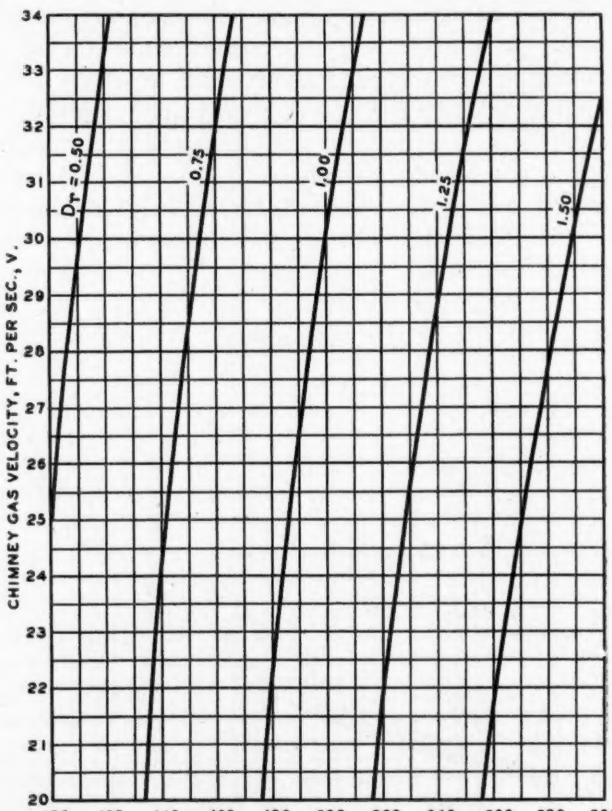


Fig. 5—Values of H for various values of D_r and V_e for a constant dia. of 10 ft. as determined from equation 10.

Side view of locomotive showing pipe drop connection to locomotive blow-off cock, overhead high pressure steam, hot water filling, washing and blow-off mains, and overhead valve operating mechanism.

Direct Steaming Railroad Locomotives from Terminal Power Stations

By PAUL E. ROLL, The Texas and Pacific Railway Company, Dallas, Tex.

Substantial savings are being effected in railroad terminals by using steam from stationary plants to bring locomotives up to working pressure. This method is a comparatively recent development which, according to the author, is now being used in about thirty engine houses. Obviously it necessitates that stationary plants providing this service must be extremely dependable and must be designed to meet fluctuating load demands. In addition to fuel economy and a considerable time saving, there are a number of other advantages, all of which the author discusses in this interesting article.

The keen industrial competition of the past few years has directed the attention of several progressive railroads toward the stationary power plant as one of several means of increasing efficiency, reducing operating expenses, and improving transportation service.

Although developments in the design of the steam locomotive have continued to increase its tractive effort and improve its fuel efficiency, there has until recently been little improvement in the methods employed to produce a working steam pressure in a locomotive boiler in the enginehouse, preparatory to dispatching the locomotive in train service.

The process of raising steam pressure in the locomotive, or in the language of the enginehouse employee, "getting her hot," has consisted of filling its boiler with warm water, laying and lighting fire on the grates or lighting the oil burner, the necessary draft being produced by use of a live steam jet ejector in the stack, called the blower. The time required to obtain 125 lb. pressure in the boiler in this manner varies from 45 min. to 2 hr., depending on the temperature of filling water, capacity of the locomotive boiler, atmospheric temperature, moisture in solid fuels or temperature of liquid fuel. Steam consumed by the blower is a considerable item, averaging 3,500 lb. per hr., and has been furnished from stationary boilers. The draft secured through the use of the steam blower has been unsatisfactory, particularly if fuel is not in condition to ignite

readily, and the deficient draft has produced a very low efficiency from the fuel burned in the locomotive fire box. The noise of the stack blower and the accompanying discharge of steam, smoke and gases, has been very objectionable, and the lighting of fires in the enginehouse always constituted a serious fire hazard.

About thirty enginehouses are now equipped with direct steaming systems to improve efficiency and secure economy in fuel consumption, and to eliminate smoke nuisance, and improve working conditions in, and general living conditions near, the engine terminals.

A direct steaming system consists of piping to transmit high pressure steam direct from the stationary boiler plant to the boilers of locomotives in the enginehouse, and is used in conjunction with the usual boiler washing plant installation. In the operation of the direct steaming system, the fire in the locomotive is extinguished or dumped and the grates and ash pan cleaned before the locomotive enters the enginehouse, into which it is propelled by the steam remaining in its boiler when it arrives at the locomotive terminal after bringing in its train.

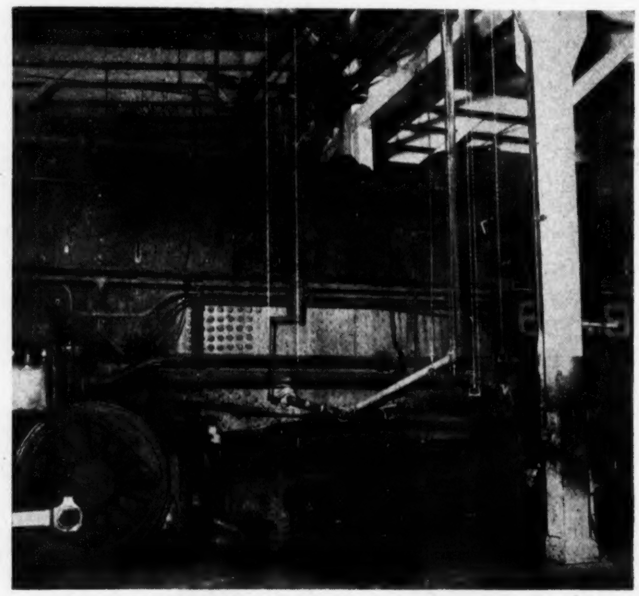
After the locomotive has been inspected and "spotted" in a stall in the enginehouse, a flexible hanging 2 in. pipe drop is connected to the locomotive blow-off cock. The flexible drop performs a combination of services, being connected through valves to overhead steam main, blow-off main and filling water main. The boiler is emptied for washout by opening the valve between the drop and the blow-off main and opening the blow-off cock on the locomotive. Steam blown off from the locomotive is utilized to heat fresh water for refilling outbound locomotives and the water blown down, after removal of sludge, is used to wash locomotive boilers.

When the boiler has been washed out, necessary repairs made and is ready to fill with water and steam, a 2 in. valve in the overhead connection to the flexible drop is opened and steam from the stationary plant is admitted to the empty boiler to heat it uniformly and avoid temperature stresses. After about three or four minutes the valve in the overhead filling water main is also opened and a mixture of live steam and hot water flows into the boiler through the flexible drop and the locomotive blow-off cock, until the water level appears at the bottom of the gage glass. The water valve is then closed and the admission of steam continued until the desired working pressure is obtained in the boiler, when the main 2 in. steam valve is closed. A ½ in. by-pass or floating valve is also provided through which constant pressure can be maintained in the boiler for an indefinite time. If it is not necessary to empty a boiler for a washout or repairs, then the floating valve is used to hold the steam in the locomotive at constant pressure until it is dispatched. If the locomotive is held for a period of several hours, by means of the floating valve, and an excess of water accumulates due to condensation, the valved connection to the blow-off main is opened and the water blown-down to the desired level. When the locomotive is ready to dispatch, all overhead valves and the blow-off cock are closed, the flexible drop disconnected, and the locomotive moved out of the house on its own steam, but fires are not lighted until it reaches the outbound spot or ready tracks.

Stationary boilers supplying steam should operate at 250 lb. per sq. in. pressure, although there are a few engine terminals operating with a pressure as low as 150 lb. with a corresponding increase in the length of time required to obtain a suitable working pressure in the locomotive boiler.

The temperature of filling water should be at least 170 deg. fahr., most modern boiler washing plants supplying filling water to the enginehouse at or above a temperature of 180 deg. fahr.

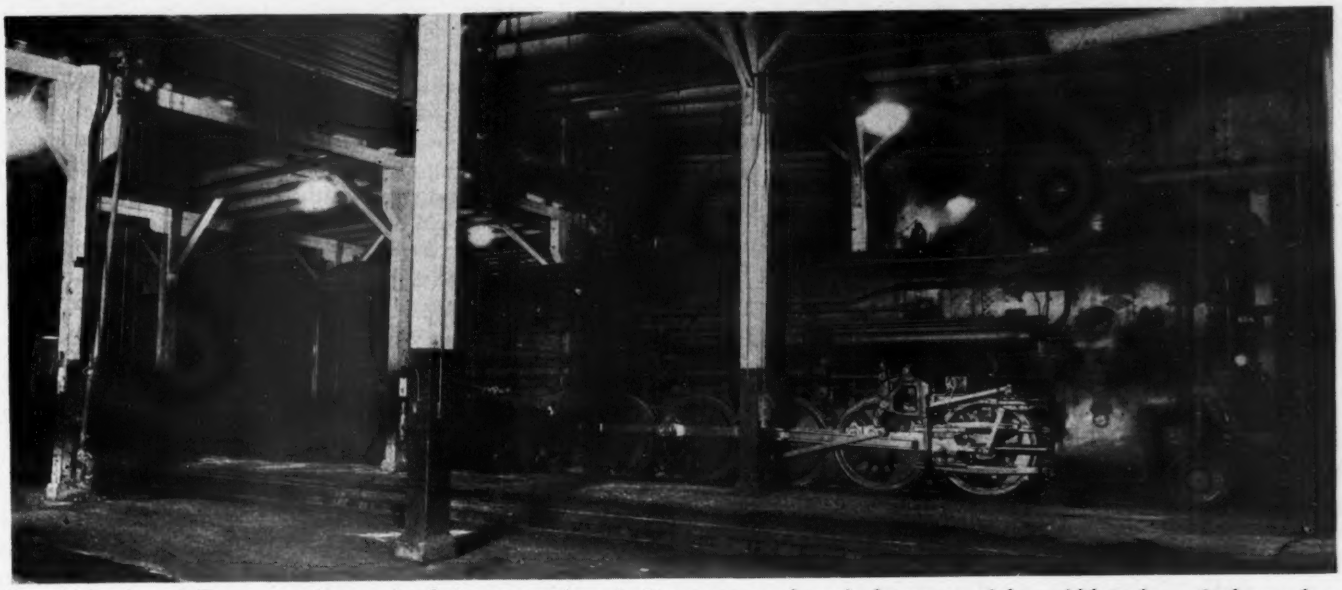
The quantity of steam and time required to fill a locomotive boiler with the direct steaming method necessarily varies at different terminals. The several factors involved are,—capacity of stationary and locomotive boilers, pressure in stationary boiler, working pressure obtained in locomotive boiler, and difference between temperature of filling water and atmospheric temperature, producing corresponding radiation losses in piping and locomotive boiler.



Side view of Texas type locomotive showing overhead direct steaming, filling and wash water, blow-off mains, combination pipe drop connected to locomotive blow-off cock, and valve operating arrangement.

Tests were conducted at the Lancaster Yard Engine Terminal of The Texas and Pacific Railway Company, where stationary boilers are operated at 225 lb. pressure, which developed that with an average of 10700 lb. of steam at an initial rate of 15000 lb. per hr., gradually decreasing as pressure in locomotive and stationary boilers equalized, an average time of 43 min. was required to bring a steam locomotive with a boiler capacity of 5000 gal., to a working pressure of 125 lb.

At the same terminal, tests were made to ascertain the quantities of fuel and time required under similar conditions to obtain a working pressure of 125 lb. in the locomotive boiler by the usual method of burning fuel oil in the locomotive firebox and using steam from the stationary boiler plant for blower stack draft, as compared with supplying steam to the locomotive boiler by the direct steaming method. The results showed that the fuel saving was 20.3 per cent and the time saved was 34.8 per cent by use of direct steaming. The estimated annual fuel saving on basis of these tests was equal to 22 per cent of the investment in direct steaming equipment and the additional stationary boiler capacity required for direct steaming.



Broadside view of Texas type locomotive being steamed up in Lancaster yard enginehouse at night. Although enginehouse has been in operation over two years, it has not been necessary to repaint white columns and roof.

The fuel saving is the result of the higher efficiences obtainable from stationary plant boilers, as compared with the relatively low efficiency of a locomotive firebox using a live steam stack blower to produce draft in the enginehouse while building up steam pressure.

In some localities, more substantial fuel savings have been accomplished by burning lower grade and cheaper fuels under the stationary boilers than it was possible to burn in locomotive fireboxes.

Direct steaming has effected other savings more difficult to evaluate such as uniformly heating all parts of the locomotive boiler and firebox water legs, thereby avoiding temperature stresses, increased utility of locomotive and enginehouse due to decreased time required for steaming, and the improved efficiency of a locomotive when dispatched with flues free from the soot which accumulates when fuel is burned in the firebox with the inadequate draft available from the stack blower.

The hazard of fire, always present when fires are lighted inside the house, is entirely eliminated. Absence of smoke and gases materially increases the life of piping and structural steel in the house and eliminates the necessity for smoke jacks and the serious objections to concrete roofs for enginehouses which otherwise exist. Better lighting is made possible by the ability to use and maintain white paint throughout the interior of the enginehouse. Workmen employed in the engine house appreciate the absence of smoke, soot, gases and noise and the generally improved working conditions and lighting and have demonstrated that their efficiency has increased in the absence of the former disagreeable conditions.

Where smoke abatement ordinances are in effect, the use of direct steaming has fully satisfied the authorities in charge of smoke prevention.

With the use of direct steaming, the power plant assumes a new responsibility and greater importance at the railroad engine terminal for it is truly the "Heart of the Terminal". The boilers must meet highly fluctuating loads and perform their service constantly or the result is delayed dispatching of motive power and disrupted train schedules.

The direct steaming systems installed are using both straight and bent water tube boilers, most of which are designed for 200 lb. to 250 lb. per sq. in. working pressure.

Coal, oil or gas fuel is being burned, the most economical fuel available at the site of the plant being used.

Practically all of the boiler installations were made especially for direct steaming as few of the older plants at engine terminals had sufficient capacity or were designed for the pressures required for direct steaming.

The employment of the system of direct steaming is just one of the many progressive forward steps that the railroads of this country have taken since the War in their effort to increase their operating efficiency, and improve their service to the American Public.

Furnace Design and Equipment for Generating Steam with Wood Refuse Fuels

By A. R. SMITH

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The author's previous article on the utilization of wood refuse fuels for steam generation, published in October COMBUSTION, dealt with the nature of wood refuse fuels, the established methods of preparation and moisture removal, and the economy of hog fuel as compared with coal and oil. The present article describes the different types of equipment used for feeding and burning wood refuse and discusses furnace, arch and grate design as influenced by the moisture content and characteristics of various wood refuse fuels.

ATTEMPTS of the past to use coal-burning stokers of conventional types to burn hog fuel have not met with success. While, in some instances, a measure of success has been reported, generally speaking, operating experiences indicate that a machine of radically different design is required. Whereas underfeed stokers have been so successful, with most classes of bituminous coals, that they have monopolized the field to the exclusion of overfeed types, the wood stokers that have met with favor have more closely approximated the latter in the essentials of their design. Since both wood and coal are solid fuels, it is of interest to note why coal stokers will not burn hog fuel in a satisfactory manner.

With a traveling grate machine, difficulty is experienced in distributing wood chips uniformly across the grate at the feed end. The result is that the fuel bed is unequal in thickness and, as a consequence, varies in its permeability to combustion air. A thin spot allows the passage of an excessive amount of air, reduces CO_2, and adversely affects combustion efficiency. Areas heavily laden have an insufficiency of air which may involve combustible losses, again reducing furnace efficiency.

Underfeed designs of the single or multiple retort type are open to the same objections as noted for traveling grates; in fact, with underfeed stokers, the maintenance of a steady and uniform feed of fuel is impossible. Both types of mechanical stokers are subject to injury from stray foreign material due to the high operative speeds required to introduce the necessary amount of a low heat value, bulky material such as hog fuel. It is to be observed that both of these types of coal stokers may be used successfully for burning wood fuel in conjunction with coal, the remarks above being applicable only to the use of wood exclusively.

A pronounced difference between wood and coal burning lies in the fact that the former is principally a surface process. The union of the combustible elements with oxygen takes place at or near the surface of the pile of wood with a large part of burning being effected in the combustion chamber above the fuel bed as a union of gases. In coal combustion, the same union of volatile combustibles with oxygen is apparent but in lesser measure. Fixed carbon, which forms a large proportion of coal, oxidizes within the body of the fuel bed.

Wood stokers separate more widely two of the functions of any mechanical stoker: first, feeding or proportioning feed to steam demand, and second, moving the fuel through the combustion chamber to the point of ash discharge. Figs. 1 and 2 show respectively representative equipment as used to feed wood and as used within the furnace to maintain an even permeable bed, to move the fuel forward through the successive stages of burning and finally to discharge the ash. Fig. 1 may be termed a feeder and Fig. 2 a stoker though both together are required to fulfil the functions of a coal stoker.

Feeding Methods

Original designs of wood burning furnaces were crude affairs into which wood was manually introduced to the grates through openings in the sides of the settings. The stoke holes remained open a large part of the time and great volumes of excess air were passed through the furnace resulting in fuel waste and damage to refractories. In addition, the V-shaped pile of fuel on the grates shielded the walls and the beneficial effect of incandescent brick-work was lost. Following this, extended furnace or dutch-oven designs were introduced with the fuel entering through one or more openings in the flat roof or arch above the grates. Combustion was improved by this change as the sides of the furnace became heated and served to drive off moisture. The hot side walls also aided in the ignition of the volatile materials resulting from the destructive distillation of wood. Following this development came the inclined grate

which made better distribution and more continuous operation possible. The lesser number of openings made the screw feeder arrangement of Fig. 1 more desirable.

Returning to a consideration of Fig. 1, note that the device consists essentially of three elements: first, a fuel storage bunker; second, a horizontal screw conveyor; and third, a star-wheel feeder. The screw-feeder receives its burden from the bottom of the bunker and moves it uniformally forward. The star-wheel device, moving in synchronism with the screw, delivers the fuel onto the grates. Agitators may be installed in the delivery spouts from the bunkers to assure free flow to the conveyor when handling wet wood chips. Such agitators may take the form of mechanical devices or may consist simply of compressed air jets so arranged as to destroy the keying of an arch of fuel formed at the bunker discharge. The entire apparatus may be arranged to operate automatically from pressure at boiler drum.

The use of feeding devices of the type described is desirable for a variety of reasons. A seal is provided between the combustion chamber and the boiler room preventing admission of air needlessly and harmfully. The seal prevents backfire into the fuel bunker

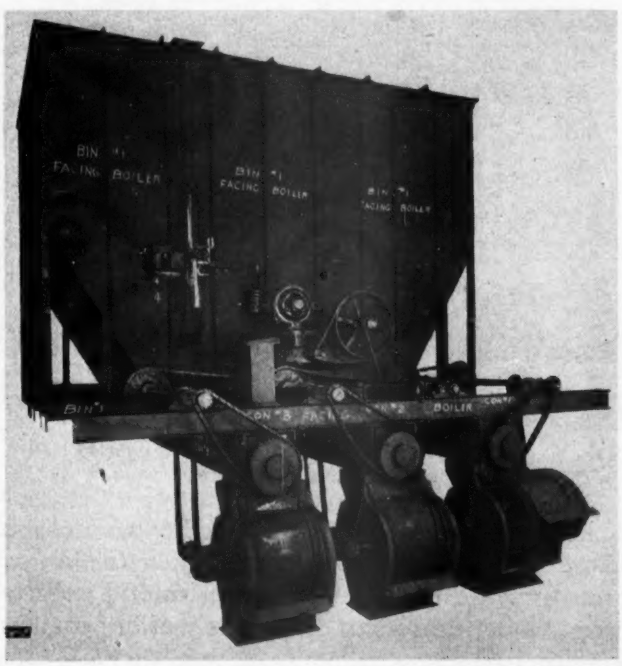

Fig. 1—View of auxiliary bins and rotary feeders which fire refuse automatically.

minimizing the hazard from fire and explosion. This type of feeder offers a steadiness of operation not obtainable by the manual introduction of fuel into an open stoke hole. It makes possible a closer control of air quantities as related to load, and lastly, reduces attendant labor in the boiler room.

Wood-waste may be transported pneumatically from a point of production to the boiler room employing a cyclone collector at the furnace to reduce the quantity of air accompanying the fuel. In this case, the cyclone is spouted directly to the furnace. The same disadvantages are present as when open stoke-hole methods are used. The amount of air required to transport the fuel is obviously far more than needed for combustion and it cannot be sufficiently reduced by the collector. The production of shavings, dust, and the like, in the factory is seldom uniform so that some provision for accumulation before the furnace, such as a storage bunker, is required if output is to be maintained uniform.

Wood Burning Furnace Design

A survey of the extensive literature of wood burning discloses a multiplicity of furnace arrangements and of combustion chamber shapes. This state of affairs can only be justified provided the nature of the wood fuel varies so that there can be no one best design, or if the physical requirements of the different types of boilers are such that variations are necessary. Both of these factors influence the problem of furnace design causing the various arrangements mentioned. With reference to the first, that of fuel variation, two distinct conditions are to be noted; first, the moisture content of the fuel, and second, the type and amount of the mineral content associated with wood-refuse. The first condition influences the shape and distribution of the refractory surfaces bounding and forming the combustion chamber, while the latter affects the design and position of furnace grates. The influence of the moisture content is so pronounced that it may be said that a furnace is designed to suit a given moisture condition, leaving out of account all other variables as of minor influence and effect.

The high hydrogen and surface moisture content of wood results in lower furnace temperatures than obtained in burning bituminous coal, lignite, or anthracite. Furnace temperature is depressed in a pronounced manner as the fuel surface moisture content increases as shown on Chart A, Fig. 2, which appeared in the author's previous article on this subject. (October issue). The chart, showing values for H (the direct measure of furnace temperature) makes calculation of furnace temperature possible and in this connection it is well to note the ignition temperatures of the various constituents of wood. Carbon requires a temperature of 900 deg. fahr., hydrogen 1130 deg. fahr., carbon monoxide 1210 deg. fahr. The moisture content must be such that a furnace temperature in excess of these values shall be attained else excessive combustible losses will occur.

Assuming the fuel moisture content to be such that adequate furnace temperatures may be attained, the designer has at hand another method for promoting complete combustion, namely, the use of the drop-nose arch or of partition walls running longitudinally of the combustion chamber dividing the latter into two or more separate chambers depending upon

the number of walls employed. No matter which is used, recourse is being had to the effect of radiant brickwork as was done in the early stages of development when combustion chamber design evolved away from the covering up of the walls with fuel, as previously discussed. The drop-nose arch is of extreme importance where moisture content is high as the arch radiates heat back to the incoming fuel, dries it, protects the fire from the cold boiler surfaces which otherwise would prematurely abstract heat, and finally, radiates to the restricted stream of gases passing beneath it raising their temperature above their ignition point.

A study of furnace temperatures at successive intervals away from a cool surface, such as a boiler or water wall, shows that the effect of absorption by the cool surface falls off rapidly as the distance away from the surface increases. The gases are to an extent impermeable to radiation, and transfer by this process is accordingly reduced as the gas stream becomes thicker. The drop-nose arch thins the gas stream to such a degree that the effect of the radiant lip is effective throughout the entire stream, the temperature of the entire mass of gas is raised to a point above the ignition temperature of the constituent substances and combustion proceeds completely. Partition walls serve the same purpose, as they are not required to radiate through so great a distance as the original bounding surfaces would be. It is at once apparent from the foregoing that the importance of radiant brickwork increases with increasing fuel moisture content.

The second important element in furnace design has to do with premature chilling of the burning volatile gases. If the active fuel bed "sees" a considerable area of cool boiler tubes, the temperature of the burning materials will be prematurely depressed and if already at or near the ignition point of the constituent requiring the highest temperature, losses will occur. The designer should take this fact into consideration and so plan the furnace as to conform to the requirements of the fuel. The higher the moisture content of the fuel the greater must be the care employed to shield the flame from the tubes of the boiler. Increasing moisture content has been shown to depress furnace temperature; hence wet wood requires extended furnaces or arches to act as shields whereas dry wood requires no precautions of this nature.

The third important consideration in furnace design is combustion chamber volume, or, to put it another way, time. Referring to the dutch-oven drop-nose arch furnace shown in Fig. 3, let us assume that the portion of the furnace before the nose has served effectively to drive off moisture at the upper zone of the grate, that it has released the volatiles further on down the grate, and finally that it has burned the fixed carbon of the fuel to CO_2 and CO at the lower portion of the bed of fuel. The initial combustion chamber has acted as a gas producer. It remains for the drop-nose arch to elevate the temperature of the gases above the ignition points of the constituents so that the combustible gases may unite with oxygen to complete combustion. If the boiler tubes followed immediately beyond the arch, combustion would cease by reason of the depression of temperature by the cold boiler tubes, and excessive losses would take place adversely affecting furnace efficiency. Such designs have been used and have been successful only because of the fortunate circumstances that the producer section was large and the wood dry. Time is required for gases to mix and combine; hence volume must be provided. From an economic standpoint, a less expensive furnace is attained by the use of primary or producer section followed by a secondary or volatile burning section.

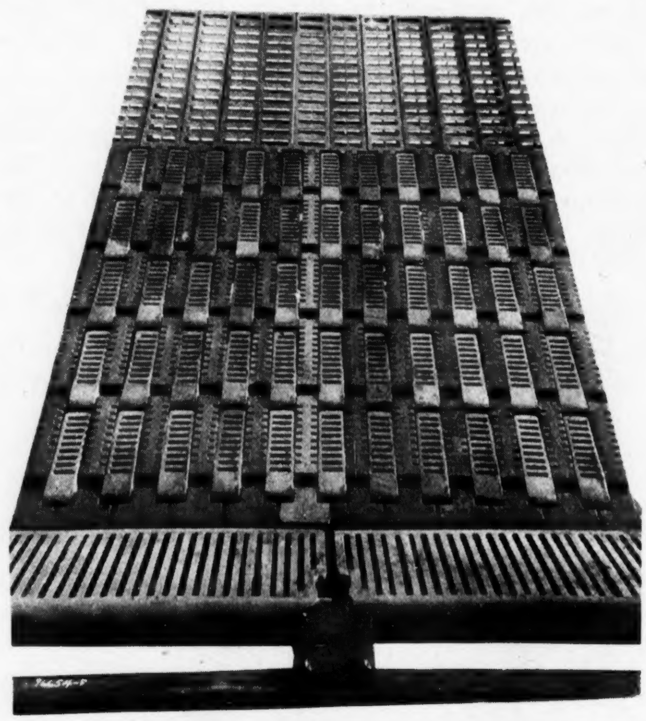

Fig. 2—Full rear view of inclined grate of a type specially designed for burning wood refuse fuels.

Arches are expensive and involve higher maintenance than a combustion chamber for which the roof is formed by the boiler so that as a rule it is better in wet-wood furnaces to divide the work of combustion into two stages.

Ideas as to the required furnace volume have gradually been revised upward in the past. Whereas a few years ago 3 to 5 cu. ft. of furnace per rated boiler hp. was deemed sufficient, experience has shown that volumes of the order of 5 to 8 cu. ft. or more give justifiably better results. Expressed in terms of heat liberation per unit of volume, as used with reference to pulverized coal forms, 3 to 5 cu. ft. per rated hp. corresponds to from 37,200 to 23,300 B.t.u. per cu. ft. when an efficiency of 60 per cent and a rating of 200 per cent are assumed. On the same basis, 5 to 8 cu. ft. corresponds to the more rational values of from

23,300 to 14,000 B.t.u. per cu ft. per hr. In making an analogy between the two fuels, however, care must be taken not to draw too heavily on the experience with one when dealing with the other, and the differences in the two problems must be observed.

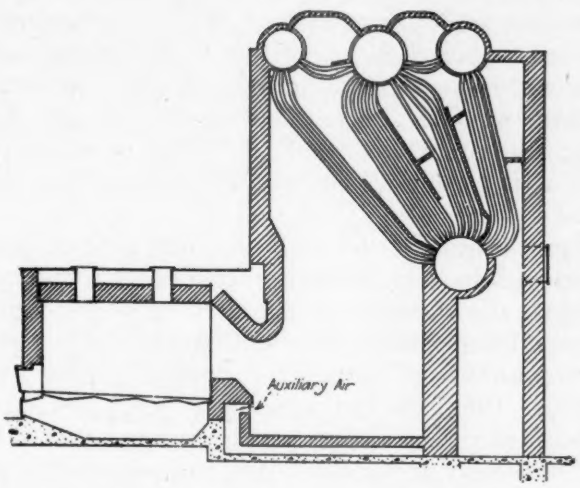

Fig. 3—Cross section of typical wet wood, drop-nose arch, flat grate furnace.

Coal contains a high ash content so that slag formation is an important criterion; the volatile content of coal is lower; furnace temperatures for a given heat liberation are higher; and lastly, the carbon content of wood fuel is not necessarily burned entirely in suspension and provision is made for its consumption on a hearth or grates. Wet wood, by reason of lower resulting temperature and consequent reduced rate of ignition and combustion, will require more furnace volume than dry wood.

The fourth and last element entering into correct furnace design is the subject of turbulence, which, expressed another way, is the proper intermixture of gas and air to prevent laning or the passage of air through the furnace uselessly and detrimentally. Turbulence is an important element with wood quite as much as in the case of pulverized-coal burning. The drop-nose arch shows an advantage in this regard in addition to those already enumerated. When the nose is so arranged as to be at a point beyond the bridge-wall, and any other location is incorrect, the gases passing under it are directed downwardly which serves to set up turbulence within the stream and beyond the arch in the secondary combustion chamber. Without this arrangement, the air would pass through the furnace uselessly as it would have no opportunity of mixing with the combustible gases. Furthermore, the air would be raised in temperature thus abstracting valuable heat from the system while the gases would carry unburned combustible to the stack; both losses would be chargeable to lack of turbulence. In a furnace of the type shown in Fig. 4, where no arches are employed, turbulence is obtained by the use of air jets. Air at high velocity enters the furnace through the jets setting up swirls and eddies which bring about the desired mixing action. In the location of air admission ports, the feature of mixing is of importance and wherever possible jets should be located with a view to promoting turbulence.

Comparison of Furnaces

In the light of the foregoing, it is of interest to compare Fig. 3 and Fig. 4. The former is a typical wet wood furnace, the latter a design well adapted to dry wood. The extremes are illustrated. Both of these furnaces may be considered as ideal insofar as they adhere to the principles already laid down.

In the wet-wood furnace, Fig. 3, a primary or producer section has been provided which affords ample incandescent refractory to dry the incoming fuel. A drop-nose arch extends beyond the bridgewall, reflecting heat back upon the fuel bed, and directs the gas stream downward setting up swirls and eddies to promote intermixture. The gases in passing under the nose are elevated in temperature and are afforded opportunity and time to burn in the amply sized, secondary combustion chamber. The arch serves to shield the fuel bed from the first pass of the bent-tube boiler. Turbulence is further promoted by the admission of air at the back of the bridgewall.

The dry wood furnace, Fig. 4, which in this instance is also designed for the use of pulverized coal,

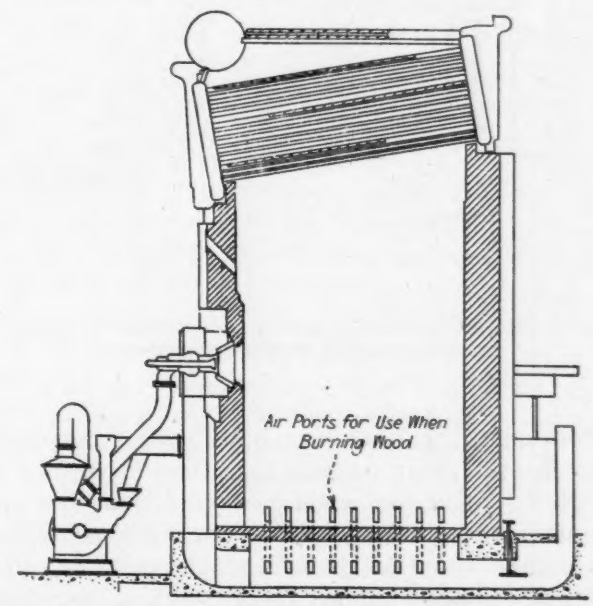

Fig. 4—Cross section of typical dry-wood burning furnace with lateral air-ports and no grates. This furnace is also designed for the use of pulverized coal.

employs no arches as the furnace temperature with dry material is sufficiently high to make this unnecessary. Wood is introduced at the front wall under the lower boiler header and falls through the furnace where a portion at least is consumed in suspension. The cross sectional area of the furnace is large so that the upward travel of gases is slow and particles are not swept out of the furnace. Ample

furnace volume is provided to afford time for combustion. Air ports are shown around the bottom of the combustion chamber at a height of about six inches above the floor. They are so arranged that they may be cleared by poking from the outside in case of stoppage. Such wood as falls to the flat floor is consumed at that point. The jets, supplied with air at about an inch water gage, set up the required turbulence.

The furnace shown in Fig. 3 will successfully burn wood with a moisture content up to 60 per cent, while that of Fig. 4 is adapted to moisture values up to about 15 per cent. For moisture contents ranging between these values, a variety of successful furnaces

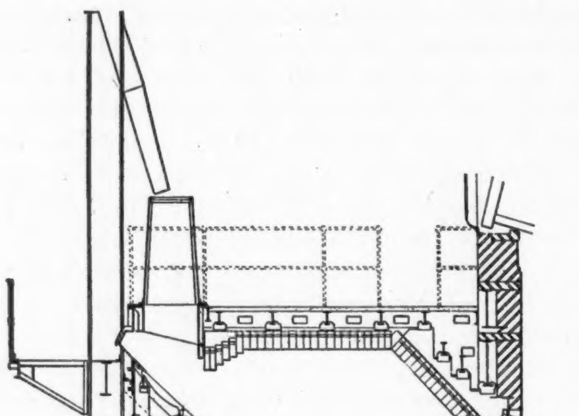

Fig. 5—Sectional elevation showing arrangement of typical inclined grate with standard grate bars.

embodying the features of each of these, in greater or lesser measure, may be designed. No one furnace is correct for all waste wood any more than any one furnace is correct for all kinds of coal. The principles above cited apply to all furnaces for wood burning and all furnace designs should adhere to these principles.

Wood Burning Grates

The surfaces provided for hog fuel to rest upon during its combustion may be conveniently divided into three general types: first, the flat grate with air openings between the bars; second; the flat hearth with ports in the sidewalls; and third, the inclined grate with air openings between the bars. The last is the most recent innovation and has pronounced advantages.

It has been indicated that the combustion of wood proceeds in three general steps: first, the contained moisture is reduced by the reflected heat of the furnace; next, the volatile constituent elements are driven off by a process of destructive distillation; and lastly, the fixed carbon content is oxidized in part to CO and in part to CO_2, the relative proportion being dependent upon circumstances. The relative amounts of fixed carbon and volatiles in wood do not vary sufficiently to affect the design of grates. Moisture content, however, again comes to the fore and it is to be noted that wet wood, by virtue of its slower combustion, requires more grate surface than dry wood.

Fig. 3 illustrates the use of the first type of grate mentioned, namely, the flat grate. Fuel is introduced through the thimbles in the arch onto the grates where it forms conical heaps and combustion proceeds at the surface of the piles. Some air passes up through the bars the remainder being taken in through auxiliary air ports. The fuel bed should be carried thicker at the rear than at the front since air entering the front is more liable to be usefully used than that entering the rear.

Fig. 4 shows the second type, namely the flat hearth, all air being taken in through sidewall ports. This type is best adapted to dry wood when the furnace is wide. The general scheme has been used with bagasse furnaces but in this application a more restricted combustion chamber is required.

The inclined grate of Fig. 5 offers several distinct advantages. Fuel is introduced at the front of the grate and fewer feed thimbles are required. Distribution of fuel over the grates is assisted by gravity and gravity also aids in moving the wood through the furnace. It is apparent that most of the air entering the furnace of a flat grate installation will enter at the edges of the conical heaps, distribution over the grate being less uniform than with the inclined grate even though a considerable number of feed openings through the arch is provided. The inclined grate may be cleaned by means of a bar while under load. The inaccessibility of the flat grate makes cleaning almost impossible without shutdown.

Associated with the ashes of wood are several compounds, such as sodium carbonate, which form, with silica, silicates of low fusion point. Silica is present in ashes, in greater or lesser measure, depending upon the nature of the refuse. Saw-mill refuse will show a high content of silica when the logs have been floated down muddy streams, the bark assuming a burden of the solid materials found in the water. Manufactury wastes have less low-fusion point, slag-forming constituent and the necessity for grate bar cleaning while under load is less pronounced. A troublesome glass-like deposit forms on the grate bars when the slag content is high which seriously clogs the air passages. Grate bar inter-

(*Continued on page 55*)

Application of the Second Law of Thermodynamics to Power Plants and Industrial Processes*

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Mr. DeBaufre's article in the November issue of COMBUSTION entitled, "Heat Engines and the Second Law of Thermodynamics," began a discussion which is continued and completed in the present article. The preceding article gave definitions of the second law which were utilized as the foundation for this branch of the subject; and a simple mathematical expression was derived to represent the law. In the present article, the author advocates a new method for applying the second law, involving a modification of the mathematical expression previously given, which permits the use of this law as a working tool in analyzing the performance of power plants and various industrial processes producing or consuming power in connection with the production or extraction of heat. The utilization of available energy and entropy in such analyses is explained.

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As mentioned in the introduction to the preceding article on "Heat Engines and the Second Law of Thermodynamics," power plants are often analyzed in accordance with the first law of thermodynamics but the second law is applied in a limited way only to steam engines and turbines and almost never to power plants as a whole.

The first law analysis accounts for each unit of energy involved in a power plant or other power producing or power consuming process; but such an analysis does not tell the whole story and may be misleading. Thus, it is often said that in a modern power plant, the boiler losses are small and the turbine losses large. This follows from the first law analysis which indicates the largest loss in a power plant to be the heat rejected to the cooling water in the condenser. From the standpoint of the second law, however, the loss in thermal efficiency of a power plant resulting from the heat rejected in the condenser is less than the loss in efficiency caused by heat transfer in the boiler from the burning fuel and the products of combustion to the water evaporating into steam. The gain in efficiency by the use of higher steam pressures is due to a reduction of this second law loss in the boiler.

A mathematical expression for the second law of thermodynamics was derived in the preceding article, namely, $(T_1-T_2)/T_1$; where T_1 is the absolute temperature of a source from which heat is withdrawn by a perfect heat engine operating in a complete cycle between that temperature and the absolute temperature T_2 of a "source of cold" to which heat is rejected by the perfect engine.

It was pointed out that this relation is of limited use because it can be applied only in cases where the source of heat remains at a constant temperature during the withdrawal of the heat which is utilized in the perfect heat engine. While this is true for the condensation of steam and other pure vapors under constant pressure, most sources of heat, such as the products of combustion in furnaces, change in temperature as heat is withdrawn from them.

In order to apply the mathematical expression for the second law to cases where the temperature varies as heat is withdrawn, the above expression must be modified by substituting for the finite quantity of heat Q the infinitesimal quantity dQ withdrawn while the temperature may be considered to remain momentarily constant. The corresponding amount of work done will then be represented by dW instead of W. The variable absolute temperature at which heat is withdrawn from the source will be represented by T. The absolute temperature at which heat is rejected will be represented by T_0 because this temperature can generally be taken as constant and equal to that of the surrounding atmosphere or of the supply of cooling water to which heat is rejected.

Available and Unavailable Energy

If we define as available energy that portion of a quantity of heat dQ which can be converted into mechanical work dW by a perfect heat engine operating in a complete cycle between the more or less variable temperature T at which the heat is withdrawn from the source and the constant temperature T_0 at which heat is rejected, then

$$\text{Available energy, A dW} = \frac{T - T_0}{T} dQ$$

The unavailable portion dQ_0 rejected by the perfect heat engine is given by

$$\text{Unavailable energy, } dQ_0 = \frac{T_0}{T} dQ$$

The ratio dQ / T was designated as dS by Clausius, who named the quantity S "entropy" from a Greek word meaning transformation, because this quantity is important in the transformation of heat into mechanical work. Hence,

$$\text{Unavailable energy, } dQ_0 = T_0\, dS.$$

Integrating between the limits S_1 and S_2, we have

$$\text{Unavailable energy, } Q_0 = T_0 (S_1 - S_2).$$

That is, when the entropy of a source of heat changes from S_1 to S_2 as a quantity of heat Q is withdrawn therefrom, the portion $T_0 (S_1 - S_2)$ is unavailable for conversion into mechanical work by a perfect heat engine working in a complete cycle between the more or less variable temperature of the source of heat and the constant absolute temperature T_0 of the surrounding atmosphere or supply of coolng water to which this unavailable portion is rejected. The decrease in entropy of the source of heat is one factor in measuring the unavailable portion of the heat withdrawn, the other factor being the absolute temperature at which the unavailable portion is rejected. The available portion converted into mechanical work by a perfect heat engine is

$$\text{Available energy, } A\,W = Q - Q_0.$$

When the source of heat is steam or some other substance of which the entropy is known, the calculation of the available portion of the heat withdrawn is comparatively simple. Thus, let us calculate the available portion of the heat withdrawn from superheated steam at 600 fahr. under 265 lb. per sq. in. absolute pressure in condensing and cooling it to 200 fahr., the temperature of ths condenser cooling water being 70 fahr. From Keenan's Steam Tables, the total heat of the superheated steam is 1316.0 B.t.u. per lb. and of the water at 200 fahr. is 167.9 B.t.u. per lb.; the entropy of the superheated steam is 1.6411 and of the water is 0.2938. Hence,

$Q = 1316.0 - 167.9 = 1148.1$ B.t.u. withdrawn per lb. of steam.

$S_1 - S_2 = 1.6411 - 0.2938 = 1.3473$ decrease in entropy of steam

$T_0 = 70 + 459.6 = 529.6$ fahr. absolute

$529.6 \times 1.3473 = 713.5$ B.t.u. unavailable per lb. of steam

$1148.1 - 713.5 = 434.6$ B.t.u. available per lb. of steam

The maximum possible efficiency of a heat engine operating in a complete cycle by withdrawing heat from the steam of the above example and rejecting heat at 70 fahr., is $434.6 / 1148.1 = 37.85$ per cent. It may be noted that the heat withdrawn from the steam is supplied to the perfect heat engine at a temperature which varies first from 600 to the saturated steam temperature of 406 fahr., then remains constant at 406 fahr. as the latent heat is withdrawn and finally decreases to 200 fahr. If the error were made of substituting the initial absolute temperature of the steam for T_1 in the expression $T_1 - T_2 / T_1$, the maximum possible thermal efficiency would apparently be 50.02 per cent instead of the true value, 37.85 per cent calculated above.

For products of combustion and other substances of which the entropy is not tabulated, the calculation of the unavailable portion of the heat withdrawn may be based on the specific heat and the change in temperature of the source of heat. Let the specific heat-temperature relation for the particular material source of heat under consideration be $C = a + b\,T + e\,T^2 +$ etc. Then,

$$\text{Heat withdrawn, } dQ = (a + bT + eT^2 + \text{etc.})\, dT.$$

Unavailable energy,

$$dQ_0 = \frac{T_0}{T} (a + bT + eT^2 + \text{etc.})\, dT.$$

$$\text{Available energy, } A\,dW = dQ - dQ_0.$$

Integrating between the limits T_1 and T_2, we have

Heat withdrawn,

$$Q = a (T_1 - T_2) + \frac{b}{2} (T_1^2 - T_2^2) + \frac{e}{3} (T_1^3 - T_2^3) + \text{etc.}$$

Unavailable energy,

$$Q_0 = a\,T_0 \log_e \frac{T_1}{T_2} + bT_0 (T_1 - T_2) + \frac{eT_0}{2} (T_1^2 - T_2^2) + \text{etc.}$$

$$\text{Available energy, } A\,W = Q - Q_0.$$

When the specific heat C of the material source of heat may be assumed constant with a sufficient degree of accuracy for the results desired, the above relations reduce to

$$\text{Heat withdrawn, } Q = C (T_1 - T_2)$$

$$\text{Unavailable energy, } Q_0 = C\,T_0 \log_e \frac{T_1}{T_2}$$

$$\text{Available energy, } A\,W = Q - Q_0.$$

As an example of the application of these relations, let us calculate the available portion of the heat withdrawn from one pound of certain products of combustion for which the specific heat-temperature relation may be taken as $C_p = 0.235 + 0.0000215\,T$ in cooling from 2000 fahr. to 500 fahr., the temperature of the atmosphere being 70 fahr. Then,

$Q = 0.235 (2460 - 960) + 0.00001075 (2460^2 - 960^2) = 407.6$ B.t.u. withdrawn per lb. of products.

$Q_0 = 0.235 \times 530 \log_e (2460 / 960) + 0.0000215 \times 530 (2460 - 960) = 134.3$ B.t.u. unavailable per lb. of products.

$A\,W = 407.6 - 134.3 = 273.3$ B.t.u. available per lb. of products.

The thermal efficiency of a perfect heat engine operating in a complete cycle between the variable temperature of the products of combustion and atmospheric temperature is $273.3 / 407.6 = 67.05$ per cent. The error sometimes made of substituting the initial temperature of the products of combustion for T_1 in the relation $(T_1 - T_2) / T_1$ would give $(2460 - 530) / 2460 = 78.46$ per cent. instead of the true maximum of 67.05 per cent.

Reversible Processes and Entropy Change

So far, we have been dealing with reversible processes free from frictional resistances and direct transfers of heat between substances at different temperatures. We have been particularly concerned with an isolated system comprising a source of heat at absolute temperature T, a source of cold at absolute temperature T_0 and a perfect heat engine with a working medium operating in a complete cycle between T and T_0, for which we have found that

$$\frac{dQ}{T} = \frac{A\,dW}{T - T_0} = \frac{dQ_0}{T_0}$$

where dQ is the quantity of heat withdrawn from the source of heat, A dW is the available portion of this heat energy converted into mechanical work and dQ_0 is the unavailable portion rejected to the source of cold. During this operation, the entropy of the source of heat is decreased by an amount dQ / T and the entropy of the working medium is increased a like amount. Also, the entropy of the source of cold is increased by an amount dQ_0 / T_0 and the entropy of the working medium is decreased a like amount. In accordance with the above relation, the decrease in entropy of the source of heat is equal to the increase in entropy of the source of cold, so that the change in entropy of the system as a whole is zero. This is representative of reversible processes, for which the total change in entropy of all substances involved is zero.

Considering the working medium only in the perfect heat engine above, the increase in entropy dQ / T during expansion from some initial to some final condition equals the decrease in entropy dQ_0 / T_0 during compression along a different path from the final to the same initial condition. From this we may conclude that the change in entropy of a substance is the same along all reversible paths between given initial and final conditions. That is, change in entropy depends upon the initial and final conditions only of a substance, being similar in this respect to internal energy instead of to the heat added or abstracted, which depends upon the intermediate conditions.

A misconception regarding entropy is that whenever this quantity is dealt with, the second law is being applied. This misconception dates back to Clausius who selected the name entropy for S in defining this quantity by dQ/T = dS and gave dQ = TdS as the mathematical expression of his second general principle of the mechanical theory of heat. This relation can be taken as the mathematical expression of the second law when dS represents the change in entropy of a source of heat and T represents the absolute temperature at which the heat dQ is rejected by a perfect heat engine operating between T and the temperature of the source. But this mathematical expression does not represent the second law when dS is the change in entropy of a body at temperature T during the addition or abstration of a quantity of heat dQ. It is true that dQ= TdS only when there are no irreversible changes within the body under consideration, such as internal friction occurring during flow through pipes or orifices; but reversible change does not constitute the second law.

Thus, for the heat dQ added to or abstracted from a substance at absolute temperature T, we may substitute in a reversible process the product of this temperature and the change in entropy dS of the substance without thought of the conversion of heat into mechanical work or of the maximum possible work that could be performed by a heat engine operating in a complete cycle between T and some other temperature. That is, neither the first law nor the second law is being applied. If the substance experiences a decrease in volume dV under pressure P while the heat dQ is added, then we could find the increase in internal energy dU by applying the first law, namely, dU = dQ + A P dV. But the mere substitution of T dS for dQ, obtaining dU = T dS + A P dV does not change the application from the first law to the second law of thermodynamics. The conception of a complete cycle as explained in the section on free energy, is required to give this expression a second law significance.

To be reversible, a process must be conducted at the point of equilibrium so that it can be carried on in either direction without loss. Reversible processes are therefore impractical because there is no tendency for them to proceed. Actual processes must be conducted at some distance from the point of equilibrium in order that a driving force will exist to cause them to proceed at reasonable speed. Actual processes are therefore irreversible.

Irreversible Processes and Entropy Change

In actual irreversible processes, the total entropy of the system increases. Thus, in an actual heat engine, there are frictional resistances and direct transfers of heat between substances at different temperatures. In a direct heat transfer, no portion of the heat transferred is converted into mechanical work; hence, in an actual heat engine, the mechanical work produced must be less than that theoretically possible in proportion to the direct heat transfers occurring. Further, the mechanical work which may be applied to useful purposes is less than that produced by an amount equal to the work expended on frictional resistances, which convert part of the mechanical work produced back into heat at temperatures lower than that of the source of heat. Consequently, for a given amount of heat dQ supplied to an actual heat engine, the useful mechanical work dW′ will be smaller and the heat rejected dQ_0' will be larger than for a perfect heat engine. That is,

$$\frac{A\,dW'}{T - T_0} < \frac{dQ}{T} \text{ and } \frac{dQ'_0}{T_0} > \frac{dQ}{T}$$

If the working medium of the actual engine passes through a complete cycle, its net change in entropy will be zero. The increase in entropy of the source

of cold, however, will be greater than the decrease in entropy of the source of heat, so that the total entropy of the system as a whole will be increased due to the presence of frictional resistances and of direct transfers of heat between substances at different temperatures.

The increase in entropy of a substance during an irreversible change cannot be calculated from the actual heat added but must be based upon some reversible change assumed in place of the actual change. Any reversible change may be assumed since the change in entropy is the same along all reversible paths between given initial and final conditions.

Irreversible changes may be said to proceed of themselves. Thus, bodies at different temperatures tend to become of a uniform temperature by direct heat transfers between them, and bodies in motion tend to be brought to rest by frictional resistances. Goodenough proposed for the second law of thermodynamics, the statement, "No change in a system of bodies that can take place of itself can increase the available energy of the system." This statement evidently requires that a definition of available energy be first established, so that it cannot be used in place of the statement of Clausius or of Lord Kelvin to prove that the maximum possible efficiency of a heat engine is independent of the working medium. Neither is the statement of Goodenough susceptible of mathematical formulation to serve as a working tool in engineering computations.

We will therefore pass on to consider what becomes of the mechanical work usefully applied by actual heat engines. There are many ways in which this work may be usefully applied; but if we investigate these useful effects,we find that eventually nearly all of the mechanical energy utilized in them is dissipated as heat by reason of irreversible changes, a negligible amount being stored up as potential energy in raised weights or as chemical or electrical energy. Thus, there is continually in progress a so-called degradation of available mechanical energy into heat energy which becomes unavailable by being dissipated into the surroundings with a corresponding increase in entropy of the bodies concerned. Clausius generalized this conclusion in the statement, "The entropy of the universe tends towards a maximum." This statement is sometimes quoted as the second law of thermodynamics, but it serves no useful purpose either as a foundation stone for this branch of the subject or as a working tool for applying the subject. Also, the statement may be too general.

There is a tendency today to question whether what is apparently true for an isolated system on this earth is also true for the universe as a whole. Instead of the universe approaching a uniform temperature with all available energy lost by being submerged as heat energy below that temperature, there may be forces at work tending to increase the availability of energy. However, as such forces are not in evidence in processes with which engineers are concerned on this earth, every irreversible change may be considered in engineering calculations as causing a corresponding loss of available energy.

Loss of Available Energy

The magnitude of the loss of available energy for any irreversible change may be found by subtracting the available energy calculated for the conditions after the change has occurred from the available energy calculated for the conditions before the change occurred. But as available energy is calculated by subtracting the unavailable portion from the total energy withdrawn from the source of heat, it is simpler to obtain the loss of available energy from the increase in unavailable energy rather than from the decrease in available energy.

As an example of loss of available energy, consider heat transfer in a boiler from the products of combustion to the steam generated. Assume the products of combustion to be cooled from 2000 to 500 fahr. in generating steam superheated to 600 fahr. under 265 lb. per sq. in. absolute pressure from feed water at 200 fahr. In preceding examples we have found that under these conditions, each pound of products of combustion gives up 407.6 B.t.u. and each pound of steam generated absorbs 1148.1 B.t.u. Neglecting the small first law loss due to heat passing through the boiler casing to surrounding objects, 1148.1 / 407.6 = 2.817 lb. of products of combustion must be cooled for each pound of steam generated. The unavailable energy for the products of combustion was previously calculated to be 134.3 B.t.u. per lb. Hence, the unavailable energy before heat transfer was 134.3 × 2.817 = 378.3 B.t.u. per lb. of steam. The unavailable energy after heat transfer is the same as previously calculated for steam, namely, 713.5 B.t.u. per lb. of steam. The loss of available energy due to heat transfer is therefore 713.5 — 378.3 = 335.2 B.t.u. per lb. of steam generated. This corresponds to a loss of 335.2 / 1148.1 = 29.20 per cent in thermal efficiency for this portion of the heat transferred.

By increasing the steam pressure from 265 to 1400 lb. per sq. in. absolute and the superheated steam temperature from 600 to 750 fahr., it may be shown by similar calculations that the loss of available energy is reduced to 21.39 per cent in thermal efficiency instead of 29.20 per cent for the lower steam pressure and temperature. The total gain of 7.81 per cent due to decreased heat transfer loss in the boiler cannot be realized in the thermal efficiency of the power plant as a whole by reason of increased second law losses in the turbine with higher steam pressure.

It may be mentioned that the above second law losses in generating steam refer only to heat transfer in the convection banks of the boiler. Still larger second law losses occur in the furnace where greater temperature differences exist between the burning fuel and the steam being generated. Also, second law

as well as first law losses occur in the heat carried away by the products of combustion to the stack, in the heat lost by "radiation" from the boiler casing, etc.

The second law method of analysis may be applied to power consuming processes as well as to power producing processes. In any case, the perfect heat engine is used as a measuring rod at each stage of the process, taking in heat from the substance at that stage and rejecting heat at the temperature of the atmosphere or of the supply of cooling water. The decrease in available energy (increase in unavailable energy) between that stage and the next stage of the process corresponds to a loss in power produced or to a gain in power consumed by that step in the process. In no case is the fluid of the actual process assumed to be the working medium in the perfect heat engine. This distinguishes the method herein described for analyzing power plants from the usual method of utilizing the Rankine cycle as a measure of the maximum possible efficiency of steam engines and turbines.

Free Energy

The second law of thermodynamics has been applied to various problems in physical chemistry, such as change of state of substances, gaseous mixtures, liquid solutions, chemical reactions, electrolytic cells, etc., by assuming the actual irreversible physical-chemical changes to be replaced by reversible changes and considering a complete cycle of events to take place. For example, chemical reactions tend to proceed until equilibrium is reached between the various components involved. The farther the conditions are from equilibrium, the greater is the driving force tending to cause the reaction to proceed. A measure of this driving force has been found in the maximum amount of work which could be performed if the final equilibrium condition were reached through a reversible change rather than through the actual irreversible chemical reaction. The value of this maximum work is determined by application of the second law of thermodynamics which shows that at constant temperature, the external work must be the same for all reversible paths between given initial and final conditions; for if there were a difference, work could be obtained by conversion of heat drawn in from the surroundings by passing from the initial to the final condition along one reversible path and returning along another reversible path. That is, by application of the second law (as stated by Lord Kelvin) we derive the fact that the external work dW must be the same for all reversible constant temperature changes and equal to the value given by the first law relation $dU = T\,dS + dW$, or $U_1 - U_2 = T(S_1 - S_2) + W$; where $U_1 - U_2$ is the decrease in internal energy during the chemical reaction and $S_1 - S_2$ is the decrease in entropy of the reacting substances, so that $T(S_1 - S_2)$ is the heat given out and W is the external work produced as mechanical and electrical energy during the equivalent reversible change.

In addition to taking place at constant temperature, a chemical reaction may proceed under constant pressure and perform the mechanical work $P(V_2 - V_1)$ or at constant volume without performing any mechanical work. There has therefore been some confusion in the value of W to select as a measure of the driving force of the chemical reaction. Helmholtz called W for the constant volume reaction, the "free" energy in comparison with $T(S_1 - S_2)$ which he called the "bound" energy. Many writers have accepted Helmholtz's definition of free energy. But as constant pressure changes are more frequent in practice and more easily produced experimentally, Lewis and Randall have proposed giving the name "free energy" to W for the constant pressure reaction. Eucken avoids this confusion by calling W, in the constant pressure relation, "maximum work."

Clapeyron's Equation

A physical-chemical relation of particular interest to engineers is that first derived by Clapeyron for the latent heat and specific volume of saturated steam and other vapors. To derive this relation, imagine a perfect heat engine operating with a saturated vapor as a working medium through a small temperature range $\triangle T$ between an upper temperature T and a lower temperature $T - \triangle T$. Denote by L the latent heat of the vapor at T and let the working medium completely evaporate from liquid into vapor at temperature T. The heat taken in at T is then equal to L, and since the efficiency of the perfect heat engine is $\triangle T / T$, the work done will be $J L \triangle T / T$, where J represents Joule's equivalent. Let the pressure change of the saturated vapor be $\triangle P$ corresponding to the temperature change $\triangle T$. Since the pressure change $\triangle P$ is very small in comparison with the change in volume from liquid V_f to vapor V_g, the work done is approximately equal to $\triangle P(V_g - V_f)$. Hence, $J L \triangle T / T = \triangle P(V_g - V_f)$, or at the limit $dT / dP = (V_g - V_f)\,T / J L$.

References

Clausius' book on "The Mechanical Theory of Heat," translated by Browne, and books on thermodynamics by Goodenough, Zeuner and others, discuss the availability of energy and entropy. The articles on thermodynamics in Glazebrook's Dictionary of Applied Physics and in the Encyclopedia Britannica will also be found of interest.

For an example of the "Analysis of power plant performance based on the second law of thermodynamics" by the present author, see the paper on this subject in the 1925 Transactions of the American Society of Mechanical Engineers, pp. 311-327. For an exposition of thermodynamics as applied to problems in physical chemistry, see the book by Hinshelwood entitled, "Thermodynamics for Students of Chemistry," also, "Fundamentals of Physical Chemistry," by Eucken, Jette and Lamer.

Blow-Down Losses and Their Calculation

By

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What are your losses due to boiler blow-down? How can they be determined and what steps should be taken to reduce them to a minimum? Taking the concentration of soluble salts as a satisfactory criterion of blow-down requirements, the author presents simple formulas for the determination of the amount of these losses. Typical examples are given to show the use of the formulas. A later article will discuss means of reducing blow-down losses.

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SINCE mechanical energy is a direct product of the thermal quality of steam, efficiency of boiler operation must be determined upon the basis of steam quality as well as steam cost. Advancement in the mechanics of condenser, feedwater heater and economizer design has aided greatly in the conservation of thermal energy. Stokers, powdered fuel systems and CO_2 recorders have been installed all with the intent of obtaining as many B.t.u. as possible from the fuel consumed.

The obnoxious and wasteful effects of boiler scale have been so evident that science has expended a great effort in the perfection of means and methods for its elimination. In fact, scale prevention is such an outstanding feature that the thought of boiler water treatment has become almost synonymous with scale prevention. Actually, efficient feedwater treatment must be vastly more than mere scale prevention. The visible evidences of the disastrous results of corrosion, pitting and caustic embrittlement have caused these phases of boiler water correction to receive unusual attention in late years. One of the phases of boiler water correction that is not so superficially evident, but which, nevertheless, is extremely important from an economic standpoint, is the loss incurred at the blow-down valve. Of what advantage is it to obtain the utmost in thermal exchange efficiency if a large percentage of the water is to be lost?

Priming and foaming are direct results of the concentration of impurities in the boiler. All too frequently wet steam is produced with consequent loss in thermal efficiency, even though the water column shows no evidence of "wild water". The chart shown in Fig. 2 gives a clear indication of the amount of such losses. To a certain extent many plants have faced the choice of two evils. They have either had to blow down the boilers and suffer blow-down losses or allow the formation of wet steam and take the consequent losses. Then too, there is the danger of damage from carrying water into prime movers. The primary reason why this phase of boiler feedwater correction has received such comparatively little attention is because of the difficulty of using recording or integrating meters in the determination of such losses.

Before any means can be taken to correct heat losses through the blow-off valve, it is necessary that we formulate some method of determining what the losses are. Concentrometers and hydrometers are used for determining concentration occurring in the boiler. The checking of the chlorides in the feed water against the chlorides in the boiler is another method that has wide use. However, until an actual cost determination is made of these losses, the need for corrective measures is not evidenced and it is for that purpose that subsequent formulas are presented.

Suspended salts are not an accurate indication since they may lodge in the boiler as scale or as sludge. Soluble salts however are evenly distributed throughout the whole volume of water in the boiler. If we presume that the water is being treated for scale prevention with a reasonable degree of efficiency, we can take the concentration of soluble salts as the criterion of blow-down requirements. The amount of soluble salts that can be carried in any boiler without carry-over or priming is a distinctly variable factor. The type of boiler and the steam space are definite factors as is the size of the steam nozzle. The degree of fluctuation of the load and the rating at which the boiler is operated are also factors. In general we might say that from 150 to 300 grains per gal. of soluble salts can be carried in the presence of some suspended matter and from 250 to 500 grains per gal. in the virtual absence of suspended matter. The allowable concentration of soluble salts decreases very rapidly with the slow accumulation of suspended matter. Since soluble salts cannot be removed except through the blow-off valve and since suspended matter can, it is wise to make provision for the removal of suspended matter by means other than through the blowoff valve. Since the removal of solubles normally entails a distinct economic loss, it is essential to keep the rate of concentration of soluble salts as low as possible.

In the formulas presented here, the following equations are used:

- W = Total boiler feed in lb.
- V = Evaporation in lb. (A unit figure of 1000 may be used).
- S = Soluble content in makeup water.
- C = Concentration of soluble salts in boiler.
- U = B.t.u. content of feed water. (determined from Chart, Fig. 1).
- U′ = B.t.u. content of boiler water. (determine from Chart, Fig. 1).
- U″ = B.t.u. content of steam. (determine from Chart, Fig. 2).
- P = Cost per gal. of makeup water including treatment.
- F = Cost per unti of fuel. (lb. of coal or cu. ft. of gas).
- B = B.t.u. content of same unit.
- E = Efficiency of boiler.
- Cl = Chlorides in makeup water. (grains per gal.).
- Cl′ = Chlorides in boiler water. (grains per gal.).
- L = Gal. of water in boiler. (determined by manufacturer's record or cubical content by calculation).

If S is the solubles in the makeup water and C is the concentration of solubles in the boiler, then it follows that $\frac{S}{C\text{-}S}$ multiplied by the unit volume of evaporation is the volume of blow-down required to maintain that concentration if the makeup is 100 per cent. If we call this amount X, then U′-U multiplied by X gives the total heat units lost and $\frac{(U'\text{-}U)F}{TE}$ multiplied by X gives the cost of heat loss incurred in the blow-down. Also if P is the cost of treated water per gal., then $\frac{P}{8.34}$ multiplied by X gives us the cost of the water blown out. If M equals the percentage of makeup, we can combine the whole into one formula as

$$M\left[\left(V\frac{S}{C\text{-}S} \times \frac{(U'\text{-}U)\,F}{TE}\right) + \left(\frac{P}{8.34} \times V\frac{S}{C\text{-}S}\right)\right] =$$

cost of blow-down for unit evaporation V. If the amount of return condensate be a constant, as is usually the case, then it is obvious that any reduction in blow-down will produce a corresponding reduction in the value of M, since makeup must be used to replace blow-down. This will still further reduce blow-down since any reduction in makeup reduces the rate of concentration.

As an illustration, let us take a plant producing one million pounds of steam daily. The soluble salts in the makeup water are 25 grains per gal. and the average soluble concentration in the boiler is 200 grains. Let us also assume a feedwater temperature of 210 deg. fahr. and a boiler pressure of 200 lb. with a boiler efficiency of 75 per cent using 10,000 B.t.u. coal costing $3.00 per ton. Makeup is 75 per cent and water plus treatment costs $.14 per 1000 gal.

Substituting these figures for the symbols we have

$$V\frac{S}{C\text{-}S} = 1{,}000{,}000\,\frac{25}{200-25} = \frac{25{,}000{,}000}{175} = 142{,}857$$

lb. water blown down. The B.t.u. content of the water at 200 lb. pressure is 363 and that of the feed water is 180. Coal costs $.0015 per lb. Therefore

$$142{,}857 \times \frac{(U'\text{-}U)\,F}{TE} = 142{,}857 \times \frac{183 \times .0015}{10{,}000 \times .75} =$$

$$142{,}857 \times \frac{.2745}{7500} = \frac{39214.2465}{7500} = \$5.228$$ or cost of fuel wasted on blow-down if 100 per cent makeup were used.

If water and treatment costs $.14 per 1000 gal., then $P = \$.00014$ and since $V\frac{S}{C\text{-}S} = 142{,}857$ we obtain

$$\frac{P}{8.34} \times V\,\frac{S}{C\text{-}S} = \frac{.00014}{8.34} \times 142{,}857 = .00001678 \times 142{,}857 = \$2.397$$

or cost of water and treatment blown down if 100 per cent makeup were used. Since M is 75 per cent, we take 75 per cent of the totals or

$$\frac{\$5.228 + 2.397}{.75} = \$5.718$$

which is the blow-down cost for this particular case.

One of the difficulties in plant operation is the lack of data required for such determinations. In some cases, the total evaporation is unknown but the total feedwater is known. In this case use the formula $V = W\text{-}W\,\frac{S}{C}$

In other cases the makeup water may be unknown but the total evaporation may be known. In such cases, use the following formula for the determination of W. $W = V\,\frac{C}{C\text{-}S}$

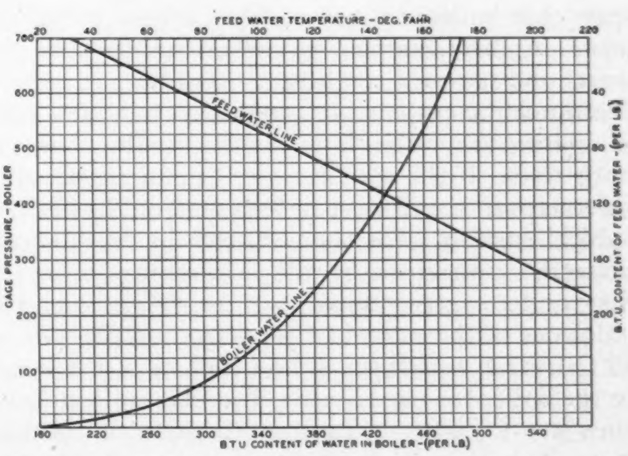


Fig. 1—Chart for determining B.t.u. content of boiler water.

If there are no convenient means for the determination of total solubles in either the feedwater or the boiler water, the values for Cl and Cl′ can be used in place of S and C. The titration for total chlorides is very simple and the reagents and directions can be

obtained from any reputable manufacturer of water treating chemicals or equipment. This method is not as satisfactory nor as accurate as the total solubles since many causes may give incorrect readings.

Very frequently no accurate data are obtainable as to the amount of makeup water used in relation to the returned condensate. There is a very simple method of obtaining this information if the tests are run

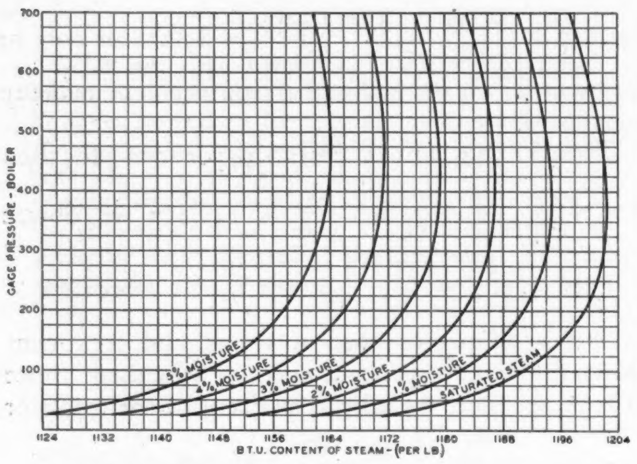


Fig. 2—Chart for determining B.t.u. content of steam with various amounts of moisture.

accurately. For this determination, two samples of water are taken from the same boiler from eight to twenty-four hours apart. These samples are taken from the water column after blowing it down to be sure that actual boiler water and not condensate is procured. There should be no blow-down between the taking of the two samples. If more than one boiler is on the line, samples will have to be taken from each boiler. Accurate chloride tests should be made on these samples as well as on the makeup water and returned condensate. The next step is to determine the exact water content of the boilers and, of course, the water level should be the same on both tests. If the manufacturer cannot supply this information, it will have to be figured on the basis of cubical content. Let the chlorides in the makeup water be represented by Cl. The chlorides in the first boiler water sample will be $Cl'(a)$ and the chlorides in the second boiler water sample will be $Cl'(b)$. If L equals the gallon content of the water in the boiler (or total boilers if several are sampled), then $L \times Cl'(a) = X$ and $L \times Cl'(b) = X'$. It follows then that $X'-X$ = total grains of chlorides added and since Cl is the grains of chloride in 1 gal. of water $\frac{X'-X}{Cl}$ = gal. of water used as makeup or

$$\frac{(X'-X)\,8.34}{Cl} = \text{lb. of water used as makeup.}$$

Therefore

$$\frac{(X'-X)\,8.34}{Cl} = \text{percentage of makeup required.}$$

If several boilers are used the values for each boiler will have to be figured and totalled. If the sample of steam condensate shows any chloride content, a correction will have to be made for this contamination.

If the steam sample shows any chloride content, a reasonably fair indication of the degree of super-saturation can be obtained by taking the ratio of the chlorides in the steam as against the chlorides in the boiler water. This will give the percentage of water in the steam and can form the means of correcting the value for U'' from the chart, Fig. 2, showing B.t.u. values for super-saturated steam. This method is not accurate and determinations should be made with a throttling calorimeter whenever possible.

Frequently, it will be found that there are no data for determining the efficiency of the boiler, efficiency, in this instance, applying only to the boiler and not to any prime movers or heat exchange equipment. When such is the case, determine the figures for U and U'' from Figs. 1 and 2. Let K equal the lb. of fuel consumed and V equal the lb. of evaporation for any unit of time. The longer the time, the less will be the factor of error. Then use the formula

$$\frac{V(U''-U)}{KT} = \text{per cent of efficiency of boiler.}$$

In making any of these calculations, corrections for the value of U will have to be made when economizers are used. Any other unusual circumstances will have to be given due consideration and corrections made where necessary. With care and diligence, it will be possible to obtain any cost data on blow-down losses with these formulas. The next step will be a consideration of means for correcting such losses and that will be dealt with in a later article.

Determination of the Economical Size of a Natural Draft Chimney

(Continued from page 24)

assumed operating conditions noted. In the absence of any operating data no more definite than that upon which the equations and curves are based, they can be used generally and, generally speaking, will give sizes which will be ample for average operating conditions around an ordinary plant.

Under no conditions should the size of a natural draft chimney be based on the horsepower of the boilers which it is to serve. Nowadays it is no more possible to determine the required size of a chimney from the boiler horsepower requirements alone than it is to select the proper size centrifugal pump from data based only on the capacity requirements or to select a fan or blower from data based only on the amount of air or gases to be moved. Just as it is necessary to know in addition the head and static pressure against which a pump and fan, respectively, are to operate so it is necessary to know in addition the head, or draft, against which a chimney is to operate.

Combined Low Temperature Carbonization and Combustion for Power Plants . . . .

A description of the new Hereng Process in France

By
DAVID BROWNLIE
L O N D O N

Low temperature carbonization of coal in conjunction with power plant operation is a decidedly interesting development that seems to be making progress in Europe. A previous article of Mr. Brownlie's described various processes of this type which have been commercially applied both in England and on the Continent. In the present article, the author describes a French development which combines the processes of coal carbonization and steam generation in a single unit. While such processes do not appear to be economically attractive in this country because of our large petroleum resources, they are, nevertheless, interesting as indicating the trend in European practice.

ONE of the inherent and very serious disadvantages of the standard steam-driven power station burning raw coal, and using condensing turbines, is the fact that the whole of the valuable by-products that can be obtained from the coal, particularly light oil, Diesel oil, and fuel oil, are burnt merely as fuel in the setting, although their value is of course very much higher per heat unit than that of the raw coal.

Prior to ten or fifteen years ago, this may not have been very important from the practical point of view because of the small size of individual stations, but the matter is very different today with plants that are, or soon will be, burning anywhere from 1,000 to 5,000 tons of coal per 24 hours. The point is that all this vast amount of coal has to be brought to the station, crushed, conveyed, stored in overhead bunkers, and discharged continuously to the combustion chambers, while the ash and clinker needs crushing, quenching, conveying, and discharging to dumps.

By a comparatively small addition of plant and equipment, it is possible to submit this coal to low temperature carbonization in front of the boiler setting, thereby recovering all the light oil, Diesel oil, and fuel oil, together with the pitch, with burning of the solid low temperature fuel and the residual gas in the boiler setting, in such amount that certainly in most countries of the world this general method is a highly important proposition. In a plant burning only 100 tons of coal per 24 hours, such an arrangement might not be profitable, but the matter is entirely different in stations consuming 1,000 to 5,000 tons per 24 hr. period. For example, by the treatment of 1,000 tons of coal per 24 hours in a combined low temperature carbonization and combustion setting for water-tube boilers, there can be recovered 18,000 gal. of low temperature tar, of which at least 50 per cent or 9,000 gal. (250 barrels) is Diesel oil, and say 2,500 gal., crude light oil, of which perhaps 1,750 gal. can be made available for use as refined gasoline.

In the United States, with its vast reserves of petroleum, the daily production from a single power station of 250 barrels of Diesel oil, 1,750 gal. of gasoline, and a large amount of fuel oil and pitch, may seem perhaps a relatively minor matter, but it is assuredly not so in the case of Great Britain, France, Germany, and other countries with little or no petroleum reserves. Also, if we are to believe the statements of the U. S. Geological Survey, this question of combined low temperature carbonization and combustion in power stations may become a proposition of the utmost importance even in the United States, as presumably the oil wells will not go on forever.

During the past fifty years, many attempts have been made to combine carbonization and combustion in steam boiler and other furnace settings using a large amount of coal, and at the present time a number of processes are in practical operation. Thus, in Great Britain, we have the Babcock or Merz & McLellan process using chain grate stokers and a vertical, mechanically-continuous retort in front of the boiler setting, internally heated by means of combustion gas from a separate gas-fired combustion chamber, using part of the residual gas from the process, mixed also with steam bled from the turbine, to give a temperature of about 1,200 to 1,290 deg. fahr., in the charge, large pieces being carbonized direct on the chain grate stoker. On similar lines in Germany is the well-known Pintsch process, using, however, as the heating medium, part of the very hot combustion gases from the boiler furnace setting. Over a dozen plants on these lines are being operated

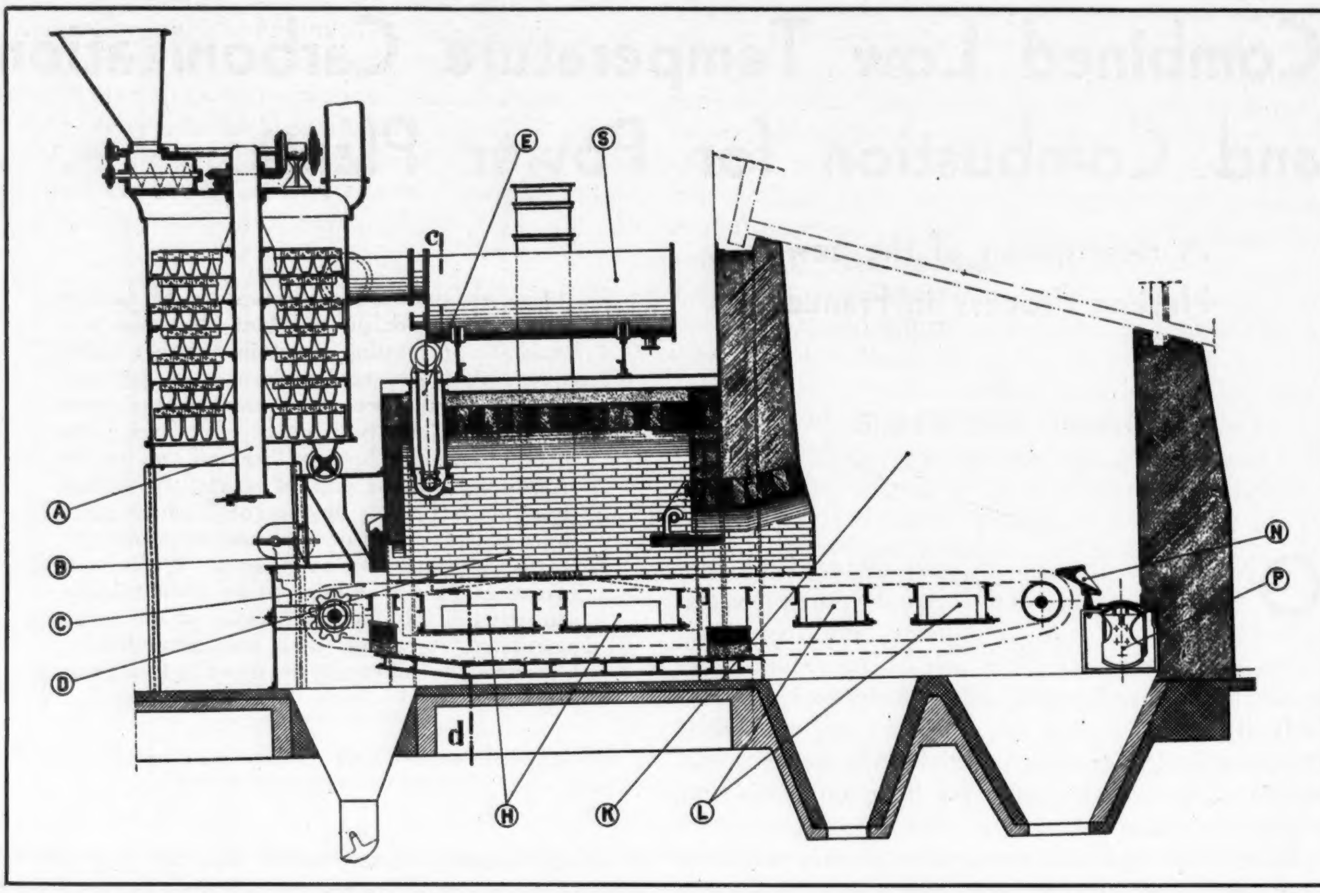


Fig. 1—Standard arrangement of the Hereng process for combined low temperature carbonization and steam generation with recovery of tar and light oil. Low temperature carbonization takes place on the first half of the chain grate stoker and the carbonized fuel is burned on the last half which is set under a water tube boiler

in Germany, Silesia, and other countries, including Norway and Brazil.

In all cases, the basic principle is to treat the mixed gases and vapors from the retort in a condensing and scrubbing plant to recover the low temperature tar and light oil before passing the gases to the combustion chamber. Also, of course, there are endless modifications possible, and, if necessary, the gas can be used separately instead of in the boiler furnace setting.

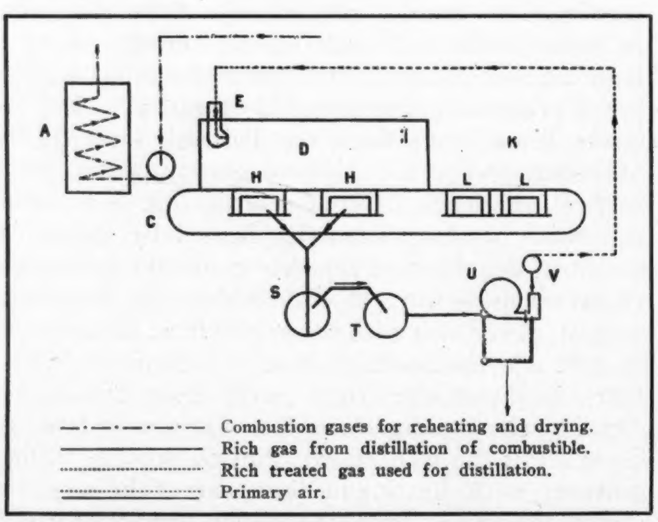


Fig. 2—Diagram indicating gas flow of arrangement shown in Fig. 1

It is the purpose of the present contribution, however, to describe particularly the new Hereng process in France, originated in 1928 by Andre Hereng, and in this connection I wish to thank M. Hereng for the detailed information he has supplied and for the new illustrations given herewith. The Hereng process is controlled by the Hydrocarb Société Anonyme, 3, Rue de Bucarest, Paris, with which M. Hereng is associated.

The basic principle of the Hereng process is combined low temperature carbonization and combustion of the raw coal on chain grate stokers, particularly in connection with water tube boiler installations. This is accomplished by greatly increasing the length of the chain grate and using the front half of the grate for carbonization purposes. The latter operation is carried out by passing stoker heated gases at about 930 deg. fahr., down through the thin layer of coal on the chain grate with all the mixed gases and vapors taken away from the bottom and delivered to the condensing plant by suction. After carbonization, the residual fuel, which of course remains on the grate, passes directly into the combustion chamber and is burnt immediately, in most cases along with the residual gas.

There is a considerable field for this type of application in France as indicated by the fact that nearly all the boiler plants of France, industrial as well as

power station, are of the water-tube type operated with chain grate stokers, and therefore the simplest method of adding low temperature carbonization is to utilize the chain grate type of apparatus. There are a number of modifications of the general design, and, if necessary, the plant can be operated without direct application to water-tube boilers. A large-scale Hereng experimental installation has been operating for a considerable time past in the neighborhood of Paris, under power station conditions and the problems are now regarded as solved.

Typical lay-outs of the Hereng process for actual large-scale operation are shown in the accompanying illustrations. Fig. 1 shows standard lay-out for operation with water-tube boilers under such conditions that the whole of the resulting low temperature fuel, and also the rich gas, is burnt in the boiler setting, while the tar and the crude light oil is recovered from the gas before combustion, the raw coal being pre-dried. The diagram, Fig. 2, shows the the path of the complete circulation of the gases.

Referring to Fig. 1, the raw coal is pre-dried at about 390 deg. fahr. in a vertical cylindrical drier *A*, operated with gearing on the mechanically-continuous principle, with a series of rabble arms, and heated internally by waste combustion gases from the chimney. The pre-dried coal passes out continuously from the bottom of this drier by way of a coal delivery valve and chute, and falls on to the long chain grate stoker *C*, equipped on the usual lines with gate *B*, adjusted vertically so as to control the amount of fuel passing.

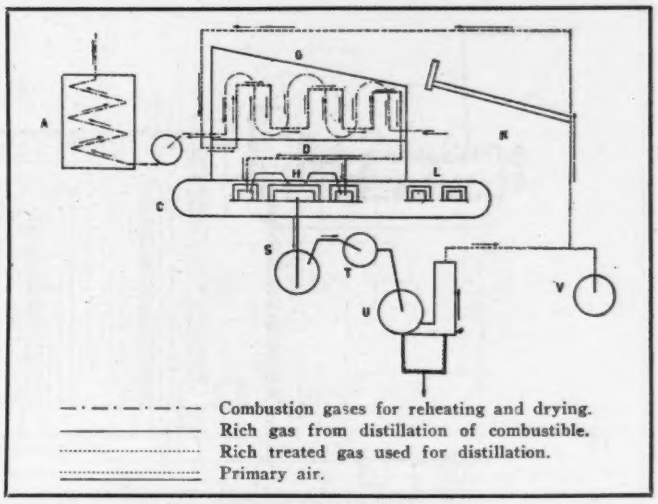


Fig. 3—Diagram indicating gas flow of arrangement shown in Fig. 4

The thin layer of pre-dried raw coal is carbonized as it travels over the first half of the chain grate which forms the bottom of a large firebrick distillation chamber *D*. The residual low temperature carbonization fuel, at a temperature of about 930 deg. fahr., passes on and is burnt in the boiler combustion chamber *K*, representing the second half of the travel of the chain grate stoker *C*. This part of the stoker is

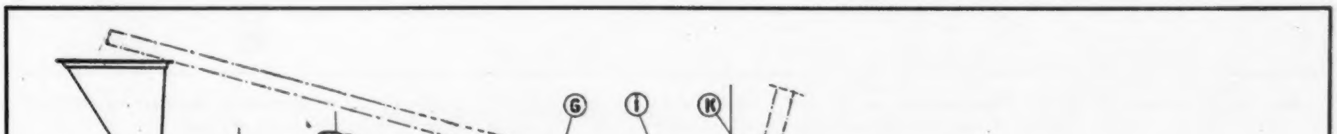


Fig. 4—Arrangement of the Hereng process for combined low temperature carbonization and steam generation. This arrangement provides for recovery of the rich gas, as well as the tar and oil, separately from the plant

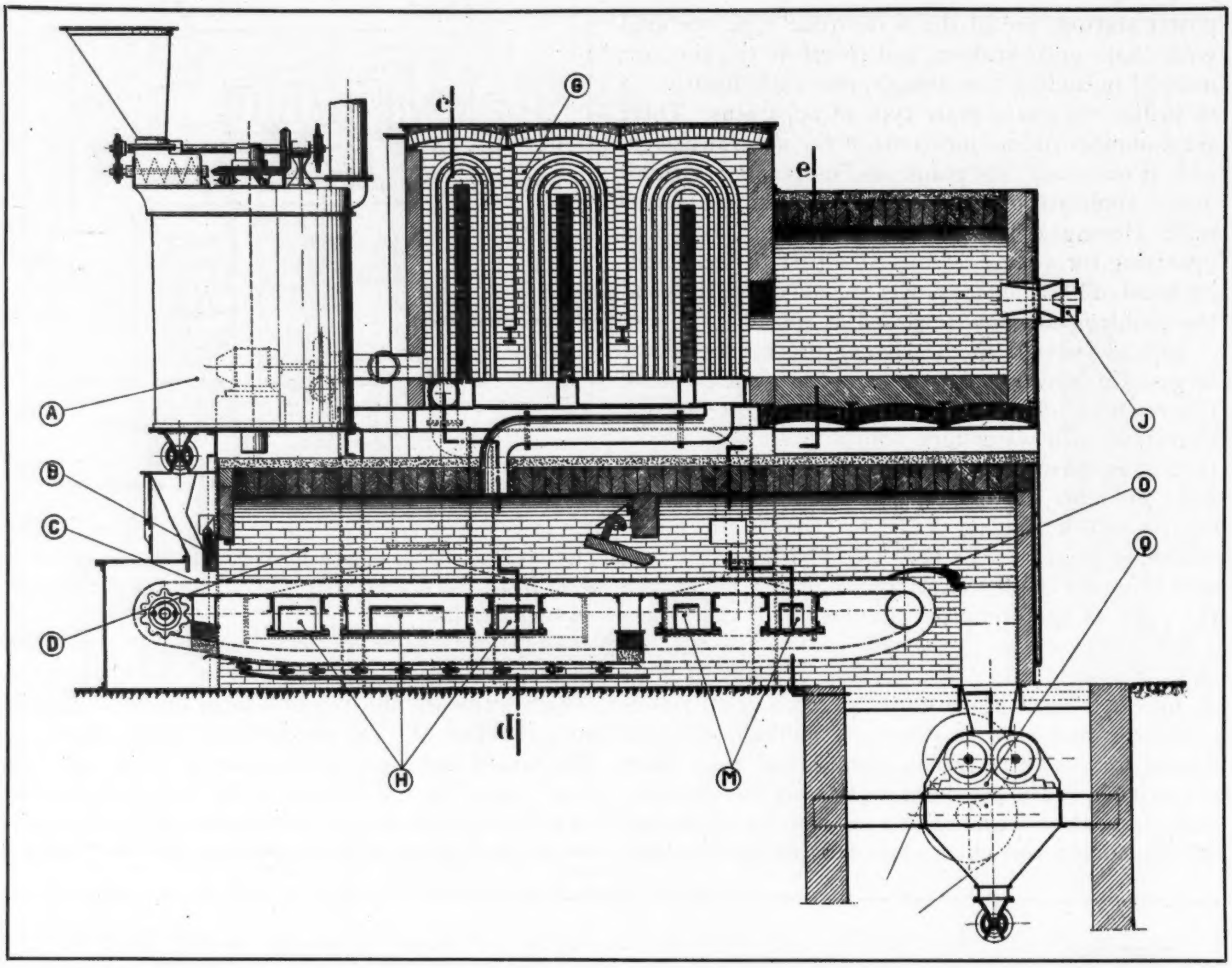


Fig. 5—Arrangement of the Hereng process for low temperature carbonization only. With this arrangement the carbonized fuel is available for sale or plant use and part of the rich gas, as well as tar and light oil, is recovered

operated with forced draft entering by means of the blast compartments L, which permit control of the amount of air. At the end of the grate, there is an arrangement N through which the ash and clinker pass as they fall to the crusher P. It will be

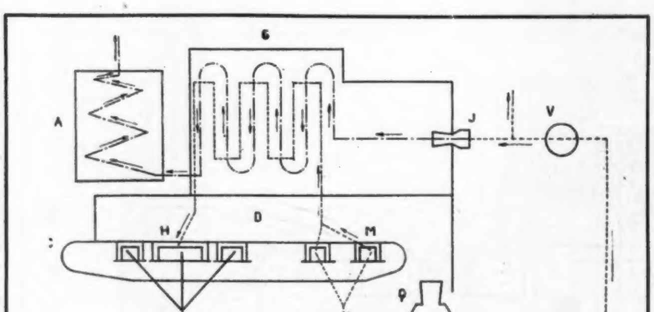


Fig. 6—Diagram indicating gas flow of arrangement shown in Fig. 5

noted there are two bottom hoppers, the one on the right for the main bulk of crushed ash and clinker and the one on the left for the siftings.

Further with regard to the carbonization, in the top of the distillation chamber D is fixed a large burner E, at which is burnt the whole, or greater part as may be required, of the residual gas after separation of the tar and the crude light oil. The combustion is so adjusted, particularly as regards air supply, that the temperature is approximately 930 deg. fahr., and the whole of these combustion gases pass downwards as indicated through the thin layer of coal as it travels on the chain grate C, thus accomplishing the low temperature carbonization. The mixed gases and vapors are taken off by means of the two compartments H beneath the chain grate stoker, which are under suction, and pass to the primary fairly high temperature condenser S (Fig. 2) for the separation of the heavy tar and also the dust. This heavy tar, it may be stated, can either be distilled separately, or, if preferable, according to circumstances, burnt in the combustion chamber D instead of, or along with, the residual gas. From the primary condenser S the gases go to a secondary main

water-cooled condenser T for the recovery of the main bulk of the tar, and following this, in the circuit, are an exhauster U and light oil scrubber V, from which latter, the final gas passes back to the burner E. Arrangements are provided for controlling the gas supply to the burners; with high-grade coal an excess of gas is generally produced which can be utilized apart from the boiler setting.

Fig. 4 shows an arrangement of plant to operate under conditions of the immediate combustion of the low temperature fuel in a boiler furnace setting but with recovery of the rich gas, as well as the tar and crude light oil, separately from the plant. Fig. 3 shows the gas circulation for this plant. As before the raw coal is pre-dried in the internally heated drier A at 390 deg. fahr., but in this case the gas used is exit gas from the recuperator G built over the top of the distillation chamber D.

The low temperature carbonization is carried out in this case by means of a current of rich gas from the process, taken after separation of the tar and the crude light oil and heated to a temperature of about 1,200 to 1,290 deg. fahr. in the recuperator setting G, through which is passed, by means of the inlet I, a considerable proportion of the very hot combustion gases from the combustion chamber K, with final discharge, as already stated, to the pre-drier A. That is, the rich gases used as the carbonizing medium are heated externally and not mixed with the combustion products, so that the final gas, in considerable surplus, having a high calorific value is available for sale, the actual carbonization, being accomplished at about 930 deg. fahr.

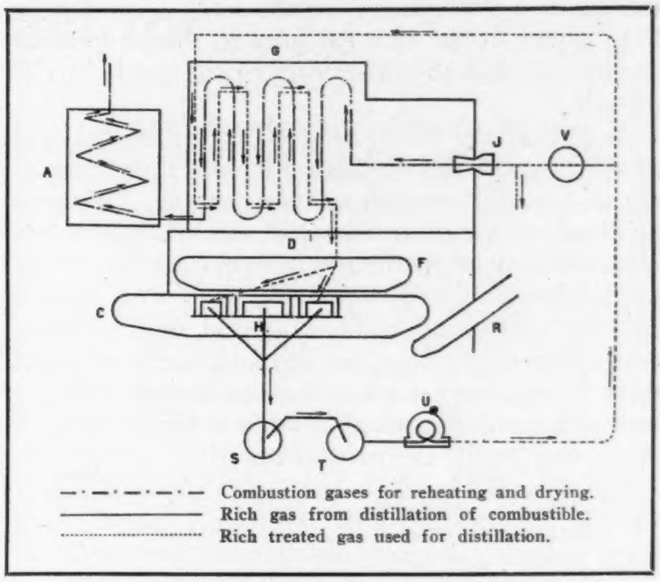


Fig. 7—Diagram indicating gas flow of arrangement shown in Fig. 8

Below the chain grate are the suction chambers $H1$, $H2$ and $H3$, connected in series to the primary high temperature condenser S (Fig. 3), for separation of the heavy tar and dust, from which the gases pass to the water-cooled condenser T for the main bulk of

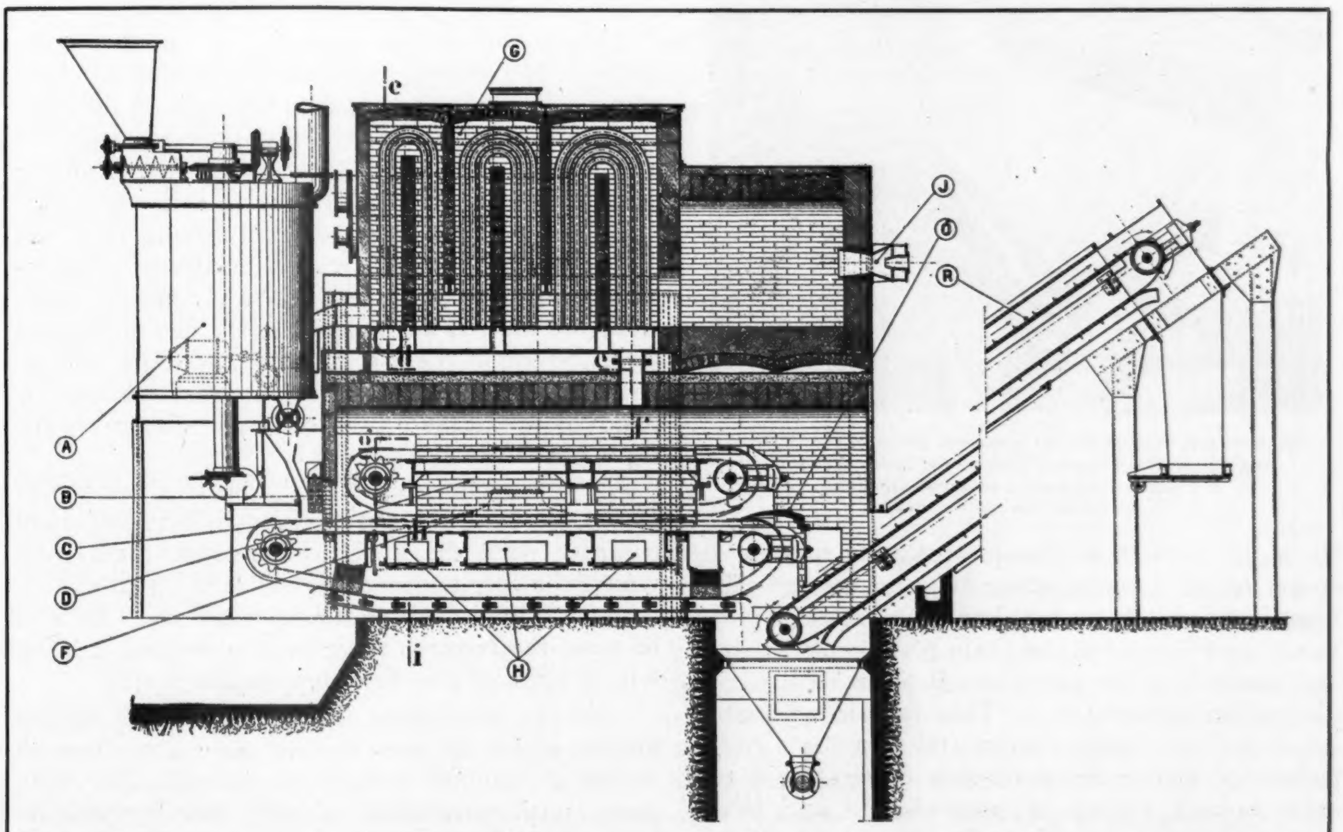


Fig. 8—Arrangement of the Hereng process for low temperature carbonization only. This arrangement involves the use of an additional upper chain grate to compress the plastic charge and provides for recovery of the carbonized fuel, part of the rich gas, tar and light oil

the tar, and the light oil scrubber *U*. As shown in Fig. 3, part of the rich gas goes to the recuperator chamber *G*, and the other part to the gas holder *V* for sale.

As already indicated, the Hereng process can also be operated as a low temperature retort only, that is, not in conjunction with a boiler setting. This plan involves the use of a somewhat shorter chain grate with recovery of all the low temperature fuel in the solid condition for sale or for use along with the rich gas. The tar and the scrubbed light oil, or alternatively of course the rich gas, can be burnt in the setting if there is no external method of utilization. Fig. 5 shows the arrangement of equipment for this type of plant, and Fig. 6, the path of the gases.

The general lay-out is similar to those of Fig. 1 and Fig. 4, consisting of the pre-drier *A*, gate *B*, chain grate stoker *C*, and long distillation chamber *D*. In this case, however, the chain grate stoker is also of considerable length because the second half of it is used to cool the low temperature fuel. The residual rich gas from the condensing plant is used for this purpose, passing down from the upper chamber on

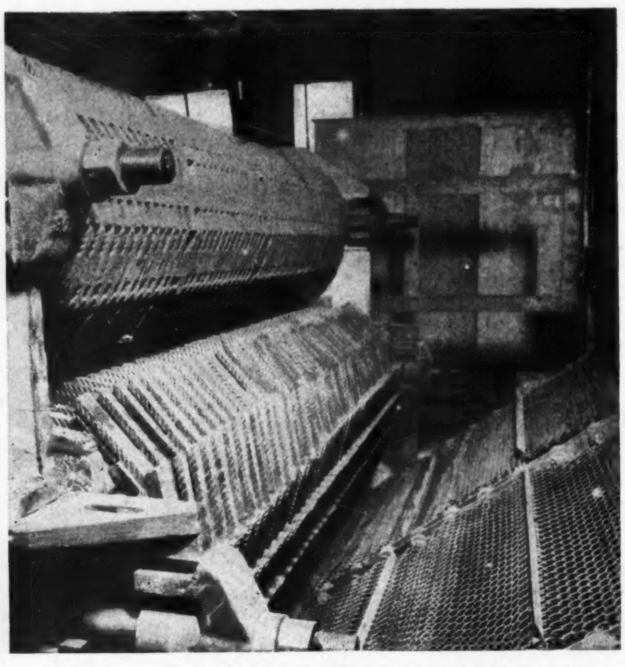


Fig. 9—Rear view of the arrangement given in Fig. 8 showing upper compressor grate and the lower carbonizing grate discharging to an inclined mechanically-continuous conveyor

the right through the low temperature fuel on the stoker to the compartments *M* under suction. The resulting, partially-cooled, low temperature fuel then passes over the end of the chain grate stoker from *D* and down into the pulverizer *Q*, from which it is discharged continuously. This fuel is admirably suited for fine pulverization and may be used as pulverized fuel under water-tube boilers or in any other furnace setting. It may also be sold in its roughly pulverized condition for use in total gasification generators.

The distillation chamber *D* is heated by part of the residual rich gas which is passed through the recuperator setting *G* where its temperature is raised to about 1,200 to 1,290 deg. fahr. It then passes down through the first part of the chain grate stoker, giving low temperature carbonization at about 930 deg. fahr., with discharge of all the mixed gases and vapors to the suction chambers *H*. The general operation is similar to the plants previously described, that is separation of the heavy tar and dust, and the main bulk of the crude light oil, with the residual rich gas partly available for sale, partly returned to the recuperator *G* to act as a heating medium, and partly consumed at the burners *J* in the end of the recuperator chamber for providing the necessary heat, the waste gases of combustion being discharged from the recuperator setting through preheater *A*.

Obviously any other convenient gas can be burnt at burners *J*, thus saving a corresponding amount of the rich gas. In fact, it may be said that the whole arrangement of the Hereng process lends itself to considerable modification according to circumstances, especially in the way of the utilization of low-grade gas such as blast furnace gas or producer gas using inferior solid fuel.

Fig. 8 shows a modification of the arrangement given in Fig. 5. In this layout also, the coal is carbonized by means of the rich residual gas which is heated in the recuperator to 1200 to 1290 deg. fahr. and then passed down through the chain grate, while the raw coal is preheated by the waste combustion gases from the recuperator setting.

This layout differs from Fig. 5 chiefly in the addition of a second, mechanically-continuous chain traveling at the same speed as the main chain grate in such a manner as to give a considerable degree of compression to the thin layer of the charge during carbonization. The claim is that in this way, when using more or less finely divided coking coal, the second or upper chain gives the maximum compression to the heated charge when in the plastic condition, and that, as a consequence, the fuel discharged from the end of the carbonization section is in relatively large pieces. The fuel in this condition is removed by means of the conveyor shown on the right of the illustration.

Fig. 9 is a photograph of the plant illustrated in Fig. 8, showing the end of the carbonization chain together with the upper compression chain from which the low temperature fuel falls on to the inclined mechanically-continuous conveyor. As will be seen, the latter is made of a close-mesh grating with a series of transverse projections or ribs.

A further interesting development of the Hereng Process which has been worked out is what may be termed a combined gasification generator and chain grate, with production of solid low temperature fuel and also producer gas, the generator being fixed at the front of the chain grate in place of the predrier.

Municipal Wastes as Fuel for Power Generation

By E. E. BUTTERFIELD, M. D.
Consulting Chemist, New York

This article presents extracts from an interesting and comprehensive paper entitled, "The Combustion of Organic Wastes with Reference to Generating Power to Meet the Power Requirements of Sewage Disposal" presented by Dr. Butterfield before the Thirty-Sixth Annual Convention of the American Society for Municipal Improvements at Richmond, Va., October 16, 1930. The original paper contains much valuable data on the subject including five tables giving typical average analyses as follows: Table 1—Composition of Garbage, as Received; Table 2—Chemical Analysis and Heat of Combustion of Garbage, Dry Basis; Table 2A—Heats of Combustion of Garbage, Observed and Computed; Table 3—Classification and Properties of Combustible Materials in Refuse; Table 4—Air Required for Combustion and Volumes of the Gaseous Products of Complete Combustion in Minimum Air.

THE use of municipal wastes as fuel for power generation has been practiced successfully in Europe and to some extent in this country. In Europe the screenings of coal ashes and the elimination of dirt and dust by screening enrich the fuel in combustible matter. In this country the tendency has been to use ashes for land-fill and to burn the rubbish and garbage alternately or in combination of the approximate proportions of two parts of raw carbage to one part of rubbish, by weight. Such a mixture is called mixed refuse in American practice. In comparing American and European figures on steam generation from municipal refuse, it is necessary to bear this difference in mind and to ascertain in each case the nature and composition of the refuse burned. In examining American results, it is necessary to have a thorough knowledge of the properties of garbage and rubbish and of the proportions in which they are available.

Heating Values of Refuse Materials

In comparing the heats of combustion of different samples of garbage, it is a safe and logical procedure to follow the methods of food and nutrition chemistry. The heats of combustion and the elementary composition can be computed from the determined proportions of protein, fat and carbohydrate just as accurately as they can be determined on duplicate samples of the same material. Given an elementary analysis of a possible fuel, there is an unfortunate tendency among combustion engineers and fuel chemists to attempt to apply the Dulong formula or the Walker rule to calculated heats of combustion. Owing to the different types of carbon linkages and the widely different heats of formation of protein, fat and carbohydrate, neither the Dulong formula nor the Walker rule can be applied to all three types of compounds or mixtures of the same. The following examples will illustrate this point:

Substance	Elementary Analysis					Heats of Combustion		
							Calculated	
	C	H	O	N	S	Determined	Dulong	Walker
Albumin	52.5	7.0	23.0	16.0	1.5	10,440	10,256	11,285
Gliadin	52.7	6.9	21.7	17.7	1.0	10,330	10,311	10,802
Butter-fat	75.0	11.7	13.3	....	...	16,740	17,111	17,439
Cellulose	44.4	6.2	49.4	....	...	7,560	6,494	7,609

The Dulong formula applies chiefly to coals high in carbon and low in oxygen, and the Walker rule applies to woods and cellulose, but the Dulong formula fails as utterly on fats and carbohydrates as does the Walker rule on protein and fats, hence it is unreasonable to apply either formula to food materials, garbage or refuse which contains all three classes of substances.

Net Heat of Combustion

In references, thus far, use has only been made of the heat of combustion as determined in the bomb calorimeter, a proportionately reduced figure for substances containing water and derived values from comparison with direct determinations. The heat of combustion as determined in the bomb calorimeter is not the heat of combustion of a substance in a furnace, for the simple reason that the heat of condensation of water formed in the combustion is measured in the bomb calorimetric determination, and that heat is not available in furnace combustion where the temperature of the furnace in operation is always above the dew-point.

It is possible to distinguish three perfectly defined and practical values:

1. Total Heat of Combustion,
2. Net Heat of Combustion,
3. Net Available Heat of Combustion.

A fourth state, a proportionate heat of combustion derived from the proportion of dry substance in a material and the heat of combustion of the dry substance, is frequently used and has been used here for comparisons. For the purpose of setting up heat balances and computing boiler efficiencies, however, the use of a proportionate heat of combustion for the material, as received, is fundamentally wrong and leads inevitably to wholly fallacious results with fuels of high moisture content.

(1) The total heat of combustion is the heat developed in the bomb calorimeter, that is, by com-

(*Continued on page 50*)

How to Check the Performance of Economizers

By B. J. CROSS, Combustion Engineering Corporation, New York

THE efficiency of an economizer may be calculated on the basis of the ratio of heat absorbed by it to the heat delivered to it. The efficiency may also be expressed by the ratio of temperature reduction of gases effected by the economizer to the maximum possible reduction in temperature. The theoretical maximum temperature reduction is the difference in temperature between the gases entering the economizer and the water fed to it. The thermal efficiency, however, is not a fair measure of the performance of the economizer. The economizer is designed to effect a certain reduction in gas temperature and rise in water temperature and its performance should be judged by the extent to which it fulfills its design requirements. There are practical limitations in the design and application of these units and an economizer with unduly high thermal "efficiency" would probably prove to be an unprofitable investment.

The upper limit for the temperature of the water is the temperature of the water in the boiler. However, unless the economizer is designed to deliver steam a considerable margin must be allowed to prevent the formation of steam during fluctuations in water flow. The lower limit of temperature is fixed by the temperature at which condensation of moisture may occur on the economizer heating surfaces. The temperature of water fed must be above that of the dew point of flue gases in order to prevent condensation of moisture with its attendant troubles.

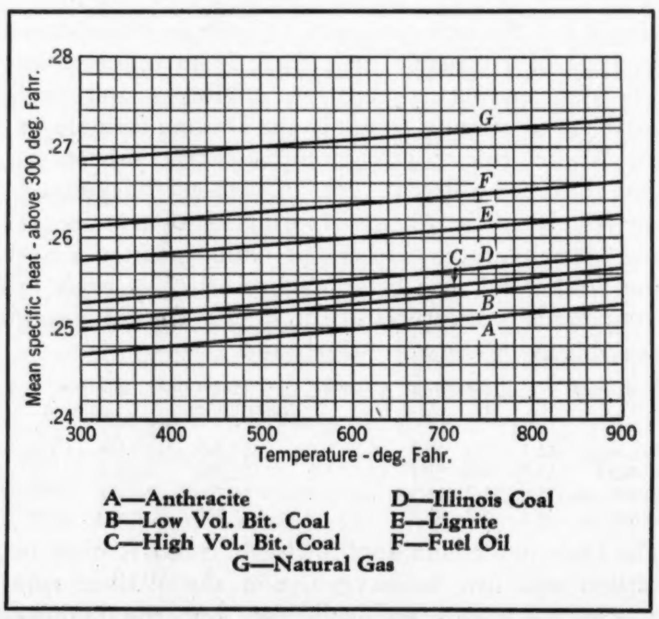
The chart on the opposite page gives the relation between water temperature rise and gas temperature drop for different ratios of gas to water. This chart may be worked from water rise to gas temperature drop or vice versa, or if both temperature differences are known, the ratio of gas to water may be determined. This chart is constructed on the assumption that all heat lost by the gas is gained by the water, the radiation loss being neglected. Thus, a water temperature rise of 95 deg. fahr. and a gas temperature drop of 262.5 deg. fahr. indicates a gas to water ratio of 1.45 when the specific heat of the gas is .25. The specific heat for water is taken as 1. The specific heat of gases is given over a range of .24 to .28 which covers the gases of combustion for all the common fuels.

The small chart, shown on this page, gives the mean specific heats for various fuels over the usual range of temperature for economizers. The constituent of the gases of combustion that chiefly affects their specific heat is the water vapor. This water vapor comes from the moisture in the fuel and from the combustion of hydrogen. The curves for the various fuels lie in the order of their hydrogen content.

If the air leakage into the economizer is negligible, the efficiency of the entire unit—boiler, economizer and air heater—may be computed, the gas temperature drop, water temperature rise and the flue gas and fuel analysis being known. Thus, in the example cited, the ratio of gas to water was 1.45. The weight of gases for a pound of fuel may be computed from the flue gas analyses as explained in previous articles of this series. If the weight of gases per pound of coal is found to be 15.5 pounds, the ratio of water to fuel is then

$$\frac{15.5}{1.45}$$

or 10.68 lb. of steam will be generated per lb. of fuel fired. This method while not recommended as a substitute for regulation tests does afford an interesting check on test data. A wide discrepancy of the results of this method and standard test results would indicate inaccuracies in temperature measurements or gas analyses.



Mean Specific Heat, Above 300 deg. fahr., of Gases, of Combustion.

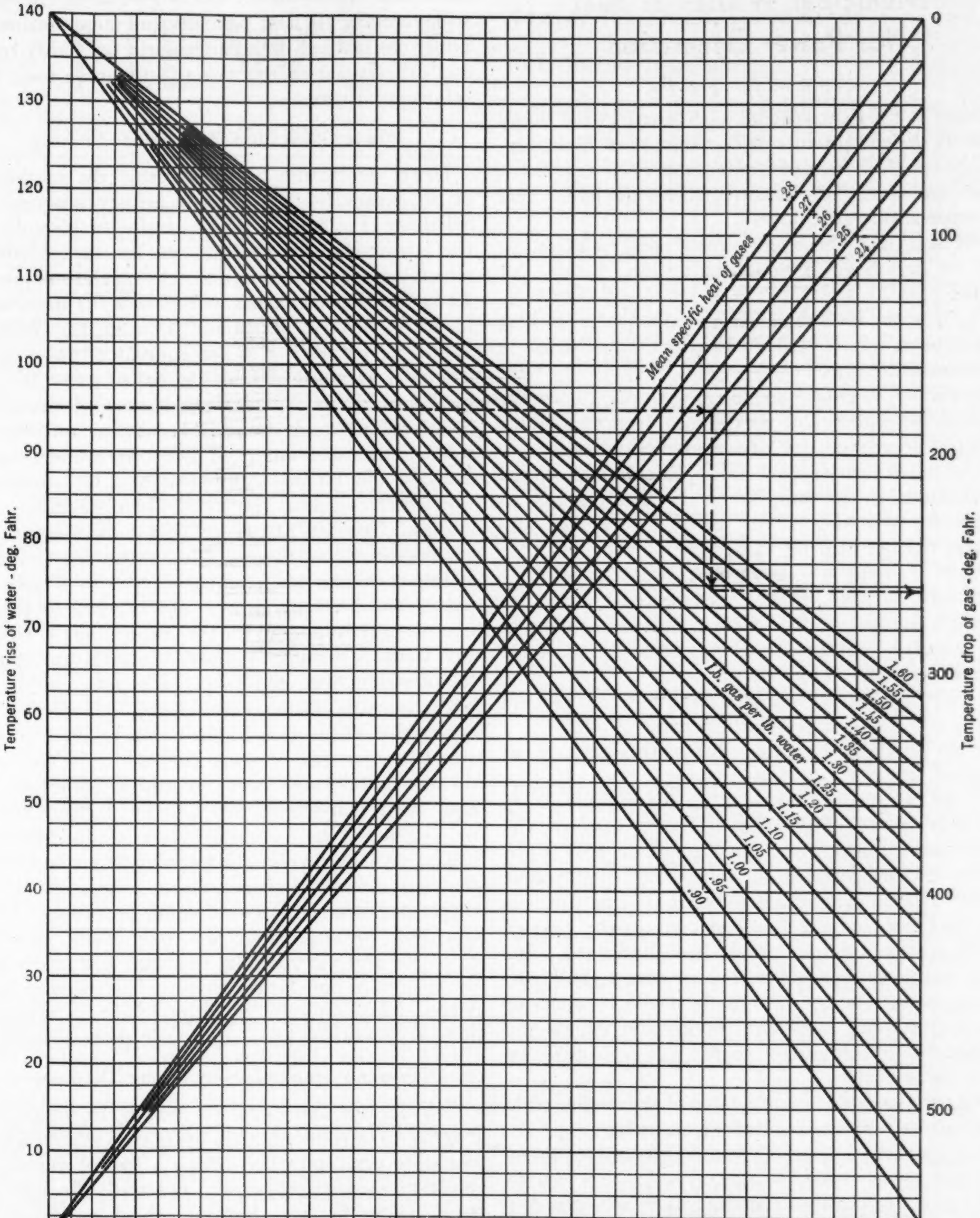


CHART SHOWING RELATION OF WATER TEMPERATURE RISE AND GAS TEMPERATURE DROP FOR ECONOMIZERS

No. 17 of a series of charts for the graphical solution of steam plant problems

Municipal Wastes as Fuel for Power Generation

(Continued from page 47)

bustion at constant volume with water in the liquid state as the end product of combustion of hydrogen.

(2) The "net heat of combustion at 68 deg. fahr., shall refer to the results corrected for the latent heat of vaporization as follows:

Total heat of combustion — 1040 (total hydrogen $\times$ 9) = net heat of combustion in B.t.u. per pound."(1)

(3) The net available heating value is the net heat of combustion corrected for hygroscopic moisture in moisture-laden fuels.

Example: Taking 8,800 B.t.u. (2) per lb. as the average weighted value for the total heat of combustion of all garbage, dry basis and the hydrogen content of the dry garbage substance, calculated from the proportions of protein, fat and carbohydrate, as 0.0667 lb. of hydrogen per lb. of dry garbage substance, the net heating value is,

$$8{,}800 - (1040 \times 0.0667) = 8{,}176 \text{ B.t.u. per lb.}$$

The same garbage in the "as received" state contained 72.13 per cent of water, and the content of dry garbage substance is therefore 27.87 per cent. The *net available heating value is,*

$$0.2787 \times [\,8{,}800 - (1040 \times 0.0667)\,] - 0.7213 \times 1040 = 1{,}528 \text{ B.t.u. per lb.}$$

The net available heating value of this garbage, as received, is

not $0.2787 \times 8{,}800 = 2{,}453$ *B.t.u. per lb.*,

which is the proportionate heat of combustion of the dry substance in the wet garbage, which heat is not available for combustion until the hygroscopic water has been evaporated and only that part of it is then available which does not contain the heat of condensation of water formed from hydrogen. Another simple calculation will show that 1,528 B.t.u. per lb. is not enough heat to heat the products of combustion with sufficient combustion air to the temperature of combustion, hence wet garbage is not a true combustible.

The net available heating value of the combustible materials in rubbish, as received, is, either,

$$7{,}257 - (1040 \times 0.062) = 6{,}696 \text{ B.t.u. per lb.,}$$

or $0.8702\,(8316 - 1040 \times 0.0472) - (1040 \times 0.071) = 6{,}736$ B.t.u. per lb. depending upon whether one takes the total combustible materials as 100 per cent cellulose, or takes the composition of dry oak wood for the composition of the ash-and-moisture-free combustible in the rubbish combustible materials.

The same method of calculation may be applied to any typical garbage or rubbish, and, in fact, to any combustible material of determined total heat of combustion and of known hydrogen and water content. The net available heating value is the value to be used in all heat balances and computations of boiler efficiencies. It is particularly foolhardy to use any other value in the case of low-grade fuels containing moisture.

(1) A.S.T.M. Serial Designation: D 271-27.

(2) Average weighted value, taken from Table 2 of paper from which this article is abstracted.

Power Generation from Refuse Burning

Power generation has been either the purpose of or an accessory to European refuse furnaces. At Salisbury, England, a refuse destructor provides all the power required for the near-by sewage works, which includes electricity for lighting and compressed air for the Shone ejectors, which now lift the sewage of the entire city. At Miami, Florida, Mr. William Sydow is generating steam equivalent to 1.14 lb. from and at 212 deg. fahr. per lb. of refuse burned. In this case only a part of the battery of furnaces is connected with the boiler, because the quantity of steam generated is sufficient to operate the sterilizers, laundry and kitchen equipment of a city hospital, for which purpose the boiler was installed. At Atlanta, Georgia, Mr. H. J. Cates, Superintendent of Incinerator Plant, is generating steam and has been selling it for the past 5 years. Sales in 1928 and 1929 passed the hundred million pound mark and were over 68 million pounds for the first 6 months of 1930. Mr. Cates is getting equivalent evaporation of 1.29 lb. of steam from and at 212 deg. fahr. per lb. of refuse burned. From these figures, as well as from other American and English equivalents, a safe generalization can be made that under present conditions of burning American municipal refuse, an equivalent evaporation of 1.1 to 1.3 lb. from and at 212 deg. fahr. can be obtained per lb. of refuse burned.

The contribution of the population to municipal refuse in American cities is from 1 lb. to 2 lb. per capita per diem. Therefore, under prevailing conditions of burning, there are potentially between 1,000,000 and 2,000,000 lb. of steam going to waste per million of population per day. This estimate is, of course, based on 1 lb. of steam per lb. of refuse, a more conservative figure than any reported results.

Power Requirements of Sewage Disposal

Calculations and estimates which are much too lengthy to introduce here indicate that when there is available per capita per diem 1.2 lbs. of municipal refuse of average properties, and with the garbage dried before furnace charging, the power requirements of the activated sludge process for average domestic sewage can be met. Further computations by C. T. Schreiber, M. E., indicate that for a population of 1,000,000 contributing approximately 2.00 lb. of refuse per capita per diem, 50,000,000 kw. hr. can be made available per annum by drying garbage with bled steam from a turbine operating with 320 lb. pressure abs. and 200 deg. fahr. superheat, bleeding 15,200 lb. of steam at 45 lb. abs. per hour and requir-

(Continued on page 55)

NEWS

Pertinent Items of Men and Affairs

Peabody Engineering Corporation Points the Way to Stabilization

ERNEST H. PEABODY, president of the Peabody Engineering Corporation, New York City made the following announcement November 7:

"Viewing with concern the tendency prevalent during periods of financial depression for the people as a whole to restrict expenditures and withhold to a large extent money which would ordinarily be placed in circulation, the Directors of the Peabody Engineering Corporation, at a meeting held November 6, at the Company's offices, Lefcourt Colonial Building, 295 Madison Avenue, New York City, declared a special dividend of fifteen dollars a share payable immediately to holders of the Common Stock of the Company.

"Also, a fund was established not to exceed $7.00 a share of Common Stock to be distributed at once among the employees of the Company in lieu of the bonus and profit sharing usually made payable during the Christmas holidays.

"This action is taken as a contribution to the efforts being made in various quarters to hasten the return to normal business conditions which at present are suffering from causes that are largely psychological.

"The customary salary increases will take effect January 1, 1931 it is announced."

•

The Bailey Meter Company, Cleveland, Ohio, Manufacturer of Automatic Combustion Control and Power Plant Metering Equipment, announces the opening of a district sales office to serve the Pacific Northwest. This office has been located in Seattle, Washington, at 406 E. 80th Street, and is under the management of L. E. Evans.

Mr. C. E. Albert has been appointed manager of the Bailey Meter Company's Kansas City office located at 1010 Coca Cola Bldg., Kansas City, Missouri. Mr. Albert was previously located at the Houston, Texas, office of the company.

•

W. D. Cameron, has been appointed manager of the Detroit office of the General Electric Company, to succeed the late J. H. Livsey, it has been announced by W. O. Batchelder, manager of the central district. Mr. Cameron has been assistant manager of the Detroit office since 1927.

Reeser Re-elected Head of Petroleum Institute

E. B. Reeser, of the Barnsdall Corporation, Tulsa, Okla., was re-elected president of the American Petroleum Institute at the annual meeting November 12. Other executive officers elected for the ensuing year were:

Vice-president-at-large—S. T. St. Clair, Union Oil Company of California, Los Angeles, re-elected; vice-president for production—Robert R. Penn, Penn Oil Company, Dallas, re-elected; vice-president for refining—R. C. Holmes, the Texas Company, New York; vice-president for marketing—E. G. Seubert, Standard Oil Company of Indiana, Chicago; executive vice-president—W. R. Boyd, Jr., American Petroleum Institute, re-elected; treasurer, Amos L. Beaty, New York, and secretary and assistant treasurer, Lacey Walker, American Petroleum Institute, New York.

The Directors approved the continuation of fundamental research in petroleum begun five years ago with gifts of $250,000 each from John D. Rockefeller and the Universal Oil Products Company. The institute directors approved a plan for raising a special fund of $100,000 a year to continue this research into the occurrence, recovery, composition and properties of petroleum. Individual companies will be asked to contribute to the fund.

•

The Ironton Fire Brick Company, Ironton, Ohio, has established a district sales and service office at 1409 Fletcher Savings & Trust Building, Indianapolis, Indiana, in charge of Carl E. Von Luhrte.

•

William Piez, of Paris, France, European Correspondent of Link-Belt Company, and a brother of Charles Piez, Chairman of the Board, died at Brussels, Belgium, on November 2nd after a week's illness.

Mr. William Piez, previous to his association with Link-Belt Company, was District Manager of the Concrete Steel Company, Chicago.

•

Roger DeWolf, formerly assistant superintendent of the Rochester Gas & Electric Corporation, has accepted the position of superintendent of steam power distribution at Reading, Pa., for the Associated Gas & Electric Company. E. R. Crofts, formerly purchasing agent of the Rochester company, will succeed Mr. DeWolf.

•

The Foxboro Company, Foxboro, Mass., manufacturer of industrial instruments has announced the appointment of A. H. Shafer as Pittsburgh District Manager to succeed H. S. Gray who has been placed in charge of office routine at the Foxboro plant.

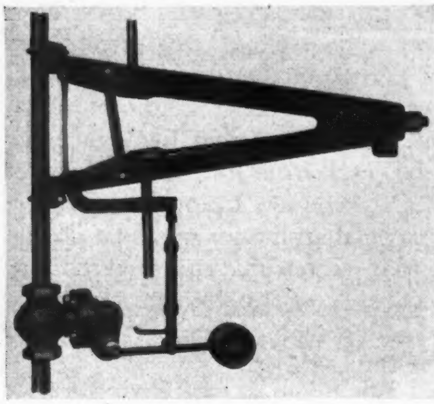
NEW EQUIPMENT

of interest to steam plant Engineers

New Feed Water Regulator

THE Copes Type OT Feed Water Regulator has been developed by the Northern Equipment Company, Erie, Pa., primarily for installation on Oil Field, Horizontal Return Tubular, Scotch Marine and other boilers of similar type. It has all the advantages of the standard Copes Regulator, plus certain distinct features which make it more practical for these particular boilers.

The Copes Type OT Regulator is simply a balanced valve actuated by an expansion



tube or thermostat. The same valve is used as in the standard Copes Regulator.

The Type OT Thermostat is made up of two short tubes connected at an angle by a heavy bronze return bend, as shown above. Both tubes are in tension and the operating effect is that of a single tube.

The thermostat is connected to the boiler in the same manner as a water column. Thus, water level in the thermostat rises and lowers in accordance with boiler water level. The upper end of the tube is filled with steam and the lower end with water. The steam, naturally, has a higher temperature than the water.

As water level in the tube rises, the average temperature of the tube decreases and it contracts. As water level lowers, the average temperature increases and it expands. This contraction and expansion acts through scientifically designed levers which multiply the effective movement of the tube to increase or decrease the amount of valve opening.

As the water level in the boiler drum rises, the valve tends to close. As the level lowers, it tends to open. Thus, the feed is according to load demands. Operation is absolutely automatic.

The regulator is particularly rugged. The thermostat is cased in malleable iron to withstand shocks and strains. In the case of portable boilers, such as are used with oil field drilling rigs, the entire unit can usually be left intact when the boiler is moved to another location.

The device is very compact. Overall length of the thermostat is less than 37 inches. Height is less than 17 inches and thickness is less than 5 inches.

Adjustment, by means of the lock nuts shown, is simple and permanent.

The regulator is easily installed. The valve may be located in the vertical feed line as shown in the illustration or in a horizontal line if desired. Screwed or flanged unions can be inserted above and below the unit to facilitate removing it from the boiler at any time this is desirable.

New CO_2 Indicator and Recorder

THE Permutit Company, 440 Fourth Avenue, New York City, has placed on the market a new model of its Ranarex CO_2 Indicator and Recorder. The well established Ranarex principle, based on specific gravity, has been retained, but the design has been simplified, and the machine made more compact and rugged.

All working parts are easily accessible. The entire indicating and recording mechanism is attached to one plate, and the driving mechanism, including the motor, to another. Both plates may be removed without the use of tools, by loosening a few wing screws; their removal opens all gas passages over their entire length. Connection between the humidifier compartments and the measuring chambers is established by channels of ample cross section.

In most instances the time lag will be less than a minute. This is of great importance to the power plant engineer and his assistants, as it enables them to correct firing conditions just as soon as a change is needed. Hence the proper percentage of CO_2 can be maintained at all times.

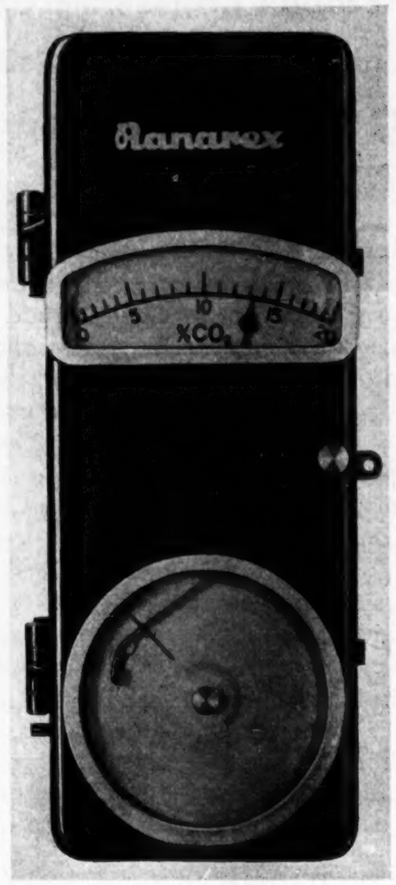
Corrosion resistant materials such as bakelite and special alloy steels are used extensively for parts in contact with flue gas. The machine is enclosed in a dust-proof cast aluminum casing and is not affected by heat, moisture or shock.

The standard 9-inch scale and circular chart 8 inches in diameter are graduated from 0 to 20 per cent CO_2, and accuracy within 0.3 per cent CO_2 is guaranteed. Scales from 0 to 30 per cent CO_2 for blast furnace gas and 0 to 40 per cent CO_2 for lime kilns are also available. The recording mechanism is driven interchangeably by a spring clock or an electric clock, as desired.

The first illustration shows the Ranarex CO_2 Indicator and Recorder—Wall Type.

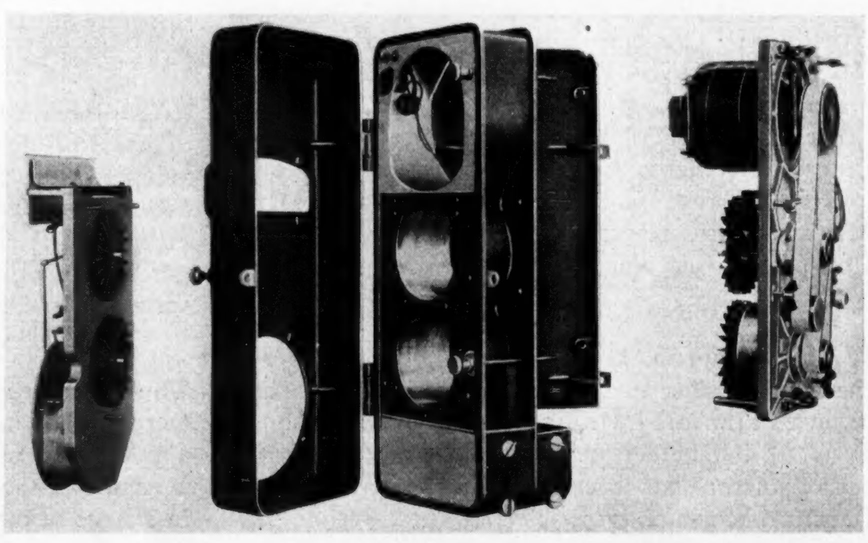
The second illustration shows the Ranarex opened for inspection. Left, plate with indicating and recording mechanism; center, dust-proof aluminum casing; right, plate with driving mechanism.

Ranarex Specific Gravity Recorders of substantially the same construction as the CO_2 recorders have been developed. The standard scale extends from 0.2 to 1.0 (air=1), but special ranges can be supplied.



A special Ranarex recorder for the control of butane-air mixing has been developed. This machine has a specific gravity range from 1.0 to 1.2 with a superimposed adjustable scale graduated in B.t.u. per cu.ft.

Ranarex machines are made with any desired scale for special applications where the specific gravity of a gas mixture is a guide to efficient and satisfactory operation.



REVIEW OF NEW TECHNICAL BOOKS

Any of the books reviewed on this page may be secured from In-Ce-Co Publishing Corporation, 200 Madison Avenue, New York

Strength of Materials

By S. Timoshenko

THE first volume of this book was reviewed in the July, 1930, issue of COMBUSTION. The second volume, which is the subject of this review, is written principally for advanced students, research engineers and designers.

The nature of Volume 2 is indicated in the preface where the author states that he has endeavored to prepare a book which contains the new developments that are of practical importance in the fields of strength of materials and theory of elasticity. Complete derivations of problems of practical interest are given in most cases. In a comparatively few cases of the more complicated problems, for which solutions cannot be derived without going beyond the limit of the usual standard in engineering mathematics, the final results only are given.

In the first chapter, the more complicated problems of bending of prismatical bars are considered. The important problems of bending of bars on an elastic foundation are discussed in detail and applications of the theory in investigating stresses in rails and stresses in tubes are given. The application of trigonometric series in investigating problems of bending is also discussed, and important approximate formulas for combined direct and transverse loading are derived.

In the second chapter, the theory of curved bars is developed in detail. The application of this theory to machine design is illustrated by an analysis of the stresses, for instance, in hooks, fly wheels, links of chains, piston rings, and curved pipes.

The third chapter contains the theory of bending of plates. The cases of deflection of plates to a cylindrical shape and the symmetrical bending of circular plates are discussed and practical applications are given. Some data regarding the bending of rectangular plates under uniform load are also given.

In the fourth chapter are discussed problems of stress distribution in parts having the form of a generated body and symmetrically loaded. Tensile and bending stresses in thin-walled vessels, stresses in thick-walled cylinders, shrink fit stresses, and also dynamic stresses produced in rotors and rotating discs by inertia forces and the stresses due to non-uniform heating are given attention.

The fifth chapter contains the theory of sidewise buckling of compressed members and thin plates due to elastic instability. In many cases, failure of an engineering structure is to be attributed to elastic instability and not to lack of strength on the part of the material.

In the sixth chapter, the irregularities in stress distribution produced by sharp variations in cross sections of bars caused by holes and grooves are considered, and the practical significance of stress concentration is discussed.

In the last chapter, the mechanical properties of materials are discussed. Such subjects as the fatigue of metals and the strength of metals at high temperature are of decided practical interest in modern machine design. These problems are treated more particularly with reference to new developments in these fields.

This book is 9¼ by 6¼ overall and contains 335 pages. The price is $4.50.

Weld Design and Production

By Robert E. Kinkead

THE author states in the preface to this book that his purpose is not only to bring together the basic information on welding, but to so arrange it that every new fact can be valued and given its proper relation to other facts. While a great deal of information has been published on the subject of welding, thus far it has not been very well organized, and it is consequently difficult to use. Accordingly, this book should serve a real need. It is written primarily for two groups, engineers who are responsible for satisfactory and safe welds, and business executives responsible for the cost element of welding jobs.

Chapters are devoted to the economics of welding, methods of welding, effect of physical conditions on weld behavior, actual welding conditions, welding procedure, control, machine welding, research and development. Charts and diagrams are used extensively throughout the text.

Generally speaking, this book is written in such a manner that it can be read intelligently by those whose information on the subject of welding is very limited. On the other hand, it covers some of the more complex phases of the subject in such a way as to be instructive even to those who have had considerable welding experience.

This book is 6 by 8½ overall and contains 108 pages. The price is $4.00.

Furnace Design and Equipment for Generating Steam with Wood Refuse Fuels

(*Continued from page 32*)

stitial openings vary from 3/8 in. to 3/4 in. depending upon the slag conditions anticipated. A wide opening is of disadvantage in that it allows fine material to fall through into the ash pit and, unless provision is made for flooding the pit, damage to the bars may result by reason of sifting fires. In general, the bar opening should be kept as small as slag formation will allow.

The ratio between boiler surface and grate area will vary, depending upon fuel moisture content and particle size. The limits are from 15 sq. ft. of boiler heating surface per sq. ft. of grate area to 40 sq. ft. of boiler heating surface per sq. ft. of grate area. Coarse fuel will require more surface than finely-hogged material by reason of the lessened rate of combustion due to the larger particle size. For attaining 200 per cent of rating with finely-hogged fuel of 40 per cent moisture content, 1 sq. ft. of grate for every 30 to 35 sq. ft. of boiler surface will be sufficient. The ratio may be increased 25 per cent when sloping grates are used.

The slope of inclined grates should exceed in angularity the angle of repose of the material to be fired so that the fuel will move forward by gravity. When the slope is insufficient to cause this forward movement, the primary purpose of the inclined grate is defeated. Slopes from 40 to 45 deg. are common. A good average figure is 42 deg. but consideration of the fuel as regards its angle of repose should determine this point.

Municipal Wastes as Fuel For Power Genearation

(*Continued from page 50*)

ing 18.44 lb. of steam per kw. hr. at the switchboard. These figures are something to pause and ponder over. Should the fuel value or the supply of refuse fall short of exact balance, there can be no objection to the use of auxiliary fuel. It is costing $1.50 and more per ton to burn municipal refuse without any return. Any temporary deficit in the public's contribution to the municipal fuel supply can be met for a fraction of the present cost of burning refuse, and the entire cost of power for sewage treatment can be reduced to a fraction of the cost of power purchased for that purpose, putting into the balance, of course, the present cost of disposing of refuse by incineration. Load factors and synchronizing power generation with power requirements are greatly facilitated by drying garbage for storing and charging as required.

NEW CATALOGS AND BULLETINS

Any of the following publications will be sent to you upon request. Address your request direct to the manufacturer and mention COMBUSTION Magazine

Boilers and Heat Exchangers

"Our Fiftieth Year" is a recently issued bulletin which outlines the growth and progress of the Henry Vogt Machinery Company during the half century of industrial activity since its founding in 1880. Following an interesting history of the company, are chapters devoted to the various types of equipment manufactured. These include bent tube boilers, box header boilers, return tubular boilers, oil refinery equipment, heat exchangers, refrigerating equipment and a complete line of drop forged steel valves and fittings. Many illustrations are included to show the diversified range of products. 16 pages and cover, 8½ x 11—Henry Vogt Machine Company, Louisville, Kentucky.

CO_2 Meter

The Brown Electric CO_2 Meter is presented in an attractive new folder. In this meter chemical solutions, rubber tubes and glassware are eliminated. There are no calculations to make—the readings are direct. As the instrument is electrically operated, it can be located at any convenient point either near the boiler or at a distance and one instrument can be used to read CO_2 at several different locations by simply switching the connections from one point to another. The meter is both indicating and recording. 12 pages, 8½ x 11 — The Brown Instrument Company, Wayne and Roberts Avenues, Philadelphia, Pa.

Feed Water Regulator

The new Copes Type OT Feed Water Regulator is described in a bulletin just issued. This regulator was developed for installation on Horizontal Return Tubular, Scotch Marine and other boilers of similar type, and consists of the standard Copes regulator valve actuated by an expansion tube device which is a modification of the standard Copes thermostat. The new regulator is rugged, compact and positive. It may be installed with the valve located in either a vertical or horizontal position. Application arrangements are illustrated. 4 pages, 8½ x 11—Northern Equipment Company, Erie, Pa.

Feed Water Treatment

"Hot Lime-Soda-Phosphate Treatment of Feed Water for High Pressure Boilers," a technical paper by C. E. Joos, Chemical Engineer of the Cochrane Corporation, has been reprinted and is available for distribution. The introduction of higher steam pressures and temperatures and greater boiler capacities has emphasized the problem of conditioning the boiler feed water. At the higher temperatures corresponding to higher pressures, overheating of the boiler metal because of scale is more likely to occur; corrosion and embrittlement are more likely to occur than at low pressures; foaming and priming are aggravated by the high rates of driving, particularly with the small drum capacities characteristic of high pressure boilers; finally, high pressure boilers cost more and are generally of larger capacity than are low pressure boilers, making it more desirable that they be held in continuous service with no shut downs on account of the condition of the water. This paper is fully illustrated by charts, diagrams and photographs. 8 pages, 8½ x 11—The Cochrane Corporation, 3160 No. 17th St., Philadelphia.

Fluid Meters

"Mechanically Operated Fluid Meters" is the title of new No. 37 bulletin devoted to Bailey Fluid Meters. This bulletin describes and illustrates various types of mechanically operated meters and auxiliary equipment for their installation. Colored meter chart records showing results obtained by the application of Bailey Meters to various processes are included, together with sections on orifice and flow nozzle location, calibration, pulsating flow, special conditions and special applications. This bulletin also includes a long list of users of mechanically operated Fluid Meters showing installation data for various fluids, kinds of service and capacities. 52 pages, 7¾ x 10½—Bailey Meter Company, Cleveland, Ohio.

High Temperature Refractories

A folder describing "Flame Brand" high temperature refractories is being distributed. This line of refractories includes a broad range of products suitable for different furnace needs such as: Laying up fire brick settings; Veneering over fire brick in furnaces; Insulating; Covering boilers and furnaces; Patching and monolithic linings. Considerable information on refractories and the use of the various materials is included. 12 pages, 4 x 8¾—King Refractories Company, 1709 Niagara Street, Buffalo, N. Y.

Natural Gas Burning Equipment

"Natural Gas as an Industrial Fuel" is the title of a new booklet which treats in comprehensive manner the use of natural gas to various industrial applications. The subjects of gas production, transmission, distribution and measurement are covered in a preliminary section which is followed by descriptions of furnaces, burners, regulators and control appliances. Typical application arrangements are illustrated and tables and engineering data related to gas fuel are included. 36 pages and cover, 6 x 9—Tate-Jones and Company, Leetsdale, Pa.

Power Transmission Chain

Union Silent and Roller Chain for power transmission is covered in a new catalog No. 200. The book is divided into two sections: Section 1 including silent chain, sprockets and casings, and, Section 2 covering roller chain and sprockets. Many application arrangements are illustrated and described and tables of sizes and prices are included. The book is a comprehensive treatise on modern high speed power transmission and the engineering data presented should prove to be of definite value to all who design or install high speed drives. 124 pages and cover, 8½ x 11—The Union Chain Manufacturing Company, Sandusky, Ohio.

Protective Coatings

"APEXIOR Coatings" applied like a paint for the preservation and protection of metal surfaces, are described in new bulletin No. 1238 just issued. It is replete with illustrations, showing a few of the many types of boilers in which NUMBER 1 is used, pictures of well-known plants which are regular users, and views of how to apply the coating together with a short description of the method of applying it to a boiler. There is also a section devoted to condensers and other cold wet applications in which the temperatures are below 125 deg. fahr. and hot dry surfaces for temperatures up to 450 deg. fahr. There is also a condensed list of users. 16 pages and cover, 8½ x 11—The Dampney Company of America, Hyde Park, Boston, Mass.

Steam Generating Equipment

A new general condensed catalog GC-6 briefly describes the complete line of fuel burning and steam generating equipment manufactured by Combustion Engineering Corporation. The apparatus shown includes both storage and direct firing systems for pulverized fuel burning, six types of mechanical stokers, bent tube boilers, sectional header boilers, box header boilers and return tubular boilers, air preheaters, water-cooled furnace walls and economizers. Numerous illustrations show details of design and application arrangements. 16 pages and cover, 8½ x 11—Combustion Engineering Corporation, 200 Madison Ave., New York.

Water Tube Boilers

Collins Western Water Tube Boilers are presented in a new catalog. The design is unique as the upper drum is placed longitudinally and the lower drum extends across the furnace. Curved tubes connect the two drums and provide a positive "ring flow" circulation. Two sizes, 91 hp. and 118 hp. are standard and interchangeable as to either a brick setting or a portable steel cased installation. With oil or gas firing, efficiencies of 80 to 82 per cent are reported for these small units even when operated up to 200 per cent rating. The design is applicable for boilers up to 5000 hp. and for pressures up to 1200 lb. 16 pages, 8½ x 11—Collins Western Corporation, Ltd., 517 Hollingsworth Building, Los Angeles, California.

NOTICE

Manufacturers are requested to send copies of their new catalogs and bulletins for review on this page. Address copies of your new literature to

COMBUSTION
200 Madison Ave., New York

Pulverized Fuel Safety Code Is Revised

A revision of the American Standard "Safety Code for the Installation of Pulverized Fuel Systems" is announced by the American Standards Association. The rapid increase in the use of pulverized fuel in many industries, resulting in new developments in methods and design of equipment, is responsible for the revision. The original code was approved by the American Standards Association in 1927.

The code covers the construction of buildings housing fuel pulverizing equipment, the ventilation of these buildings, and specifications for dust collection systems. Specifications are given covering methods of preventing explosions through static, through the drying of coal, and through its transportation through pipe lines. The code also contains suggestions for safe operating rules to be printed on instruction cards which can be used for guidance of the employees in charge of the operation of such systems.

The standard which is part of a comprehensive project for the prevention of dust explosions, was developed, and the revision undertaken, under the sponsorship of the National Fire Protection Association and the United States Department of Agriculture. The American Standards Association Technical committee in charge of the project, under the chairmanship of D. J. Price, Bureau of Chemistry, U. S. Department of Agriculture, Washington, D. C., includes representatives of the following organizations:

American Spice Trade Association; Associated Corn Products Manufacturers; Associated Factory Mutual Fire Insurance Companies; Conference of Special Risk Underwriters; Dust Collection Equipment Manufacturing Group; Grain Elevator Construction; International Association of Fire Chiefs; International Association of Fire Fighters; Association of Governmental Officials in Industry; International Association of Industrial Accident Boards and Commissions; Mutual Fire Prevention Bureau; National Board of Fire Underwriters; National Bureau of Casualty and Surety Underwriters; National Electric Light Association; National Electrical Manufacturers Association; Railway Fire Protection Association; Terminal Grain Elevator Merchants' Association; Western Actuarial Bureau; United States Bureau of Mines; United States Department of Agriculture; United States Department of Labor; Society of Grain Elevator Superintendents of North America; Pulverized Fuel Equipment Association; Underwriters' Laboratories; and a representative of the sugar industry.

Copies of the code are available for five cents each from the American Standards Association, 29 West 39th Street, New York City.

Boiler, Stoker and Pulverized Fuel Equipment Sales

Total figures to October 1, as reported to the Department of Commerce by the leading manufacturers in each industry

Boiler Sales

	Total 9 mo. 1930		Total 9 mo. 1929		Sept., 1929		Sept., 1930	
	No.	Sq. ft.	No.	Sq. ft.	No.	Sq. ft.	No.	Sq. ft.
Water tube	906	4,815,193	1339	7,692,059	144	966,656	78	565,636
H.R.T.	739	1,002,906	1102	1,458,504	115	138,759	93	119,291

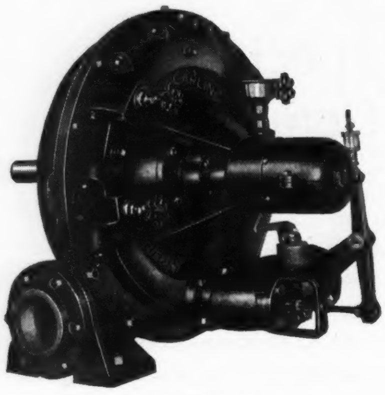
Mechanical Stoker Sales

Year and Month	Total		Type of Boiler			
			Fire-tube		Water-tube	
	No.	HP.	No.	HP.	No.	HP.
1929						
July	186	65,197	88	11,922	98	53,275
August	199	54,929	94	13,981	105	40,948
September	155	45,685	71	9,791	84	35,894
Total (9 mo.)	1,352	459,032	544	81,613	808	377,419
Total (year)	1,716	599,585	706	102,515	1,010	497,070
1930						
January	53	13,198	24	2,872	29	10,326
February	73	22,648	26	3,732	47	18,916
March	89	32,403	45	6,128	44	26,275
April	108	35,903	46	6,984	62	28,919
May	96	31,956	41	5,703	55	26,253
June	151	47,803	70	10,100	81	37,703
July	150	37,761	83	11,434	67	26,327
August	115	29,988	61	10,587	54	19,401
September	128	42,899	71	9,186	57	33,713
Total (9 mo.)	963	294,559	467	66,726	496	227,833

Pulverized Fuel Equipment Sales

Year and Month	Central System			Unit System		
	No. of Pulverizers	Total Rated capacity in tons of coal per hour	Total Rated hp. of boilers equipped	No. of Pulverizers	Total Rated capacity in tons of coal per hour	Total Rated hp. of boilers equipped
1930	For Installation Under Water-Tube Boilers					
January	1	6	1,600	52	565	59,742
February	2	20	3,000	29	175	23,305
March	2	50	6,414	16	33	9,995
April	....	....	....	31	139	37,993
May	3	80	11,360	30	196	22,625
June	1	6	802	15	28	7,146
July	2	22	1000	12	29	20,424
August	....	....	....	4	13	1,454
September	....	....	....	24	112	18,729
Total (9 mo.)	11	184	24,176	213	1,290	201,413
	For Installation Under Fire-Tube Boilers					
January	....	....	....	6	35	965
February	....	....	....	2	13	305
March	....	....	....	3	3	450
April	....	....	....	3	3	780
May	....	....	....	....	....	....
June	....	....	....	....	....	....
July	....	....	....	....	....	....
August	....	....	....	3	3	712
September	....	....	....	6	3	900
Total (9 mo.)	....	....	....	23	60	4,112

CARLING



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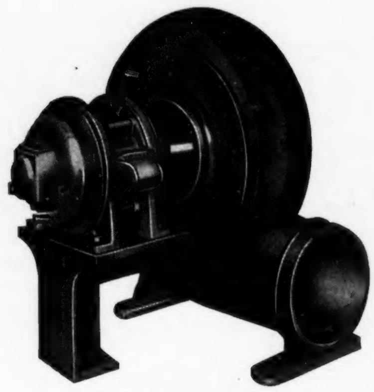

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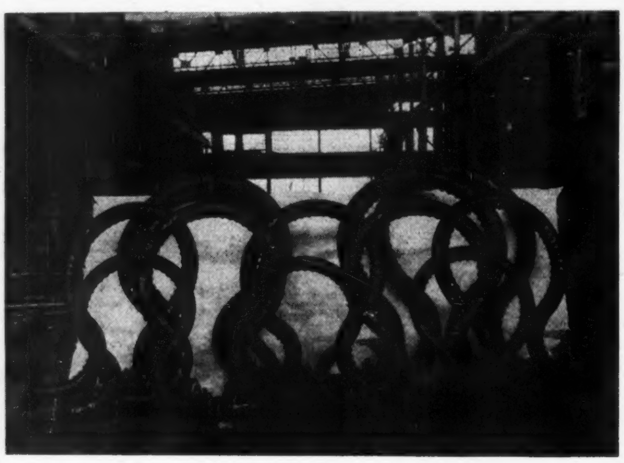

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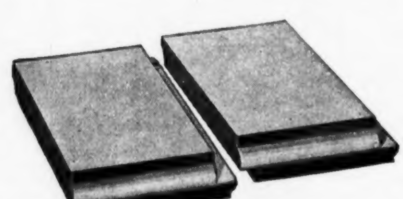

Two Design A-9 Interlocking Baffle Tiles

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